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NATO Committee on the Challenges of Modern Society

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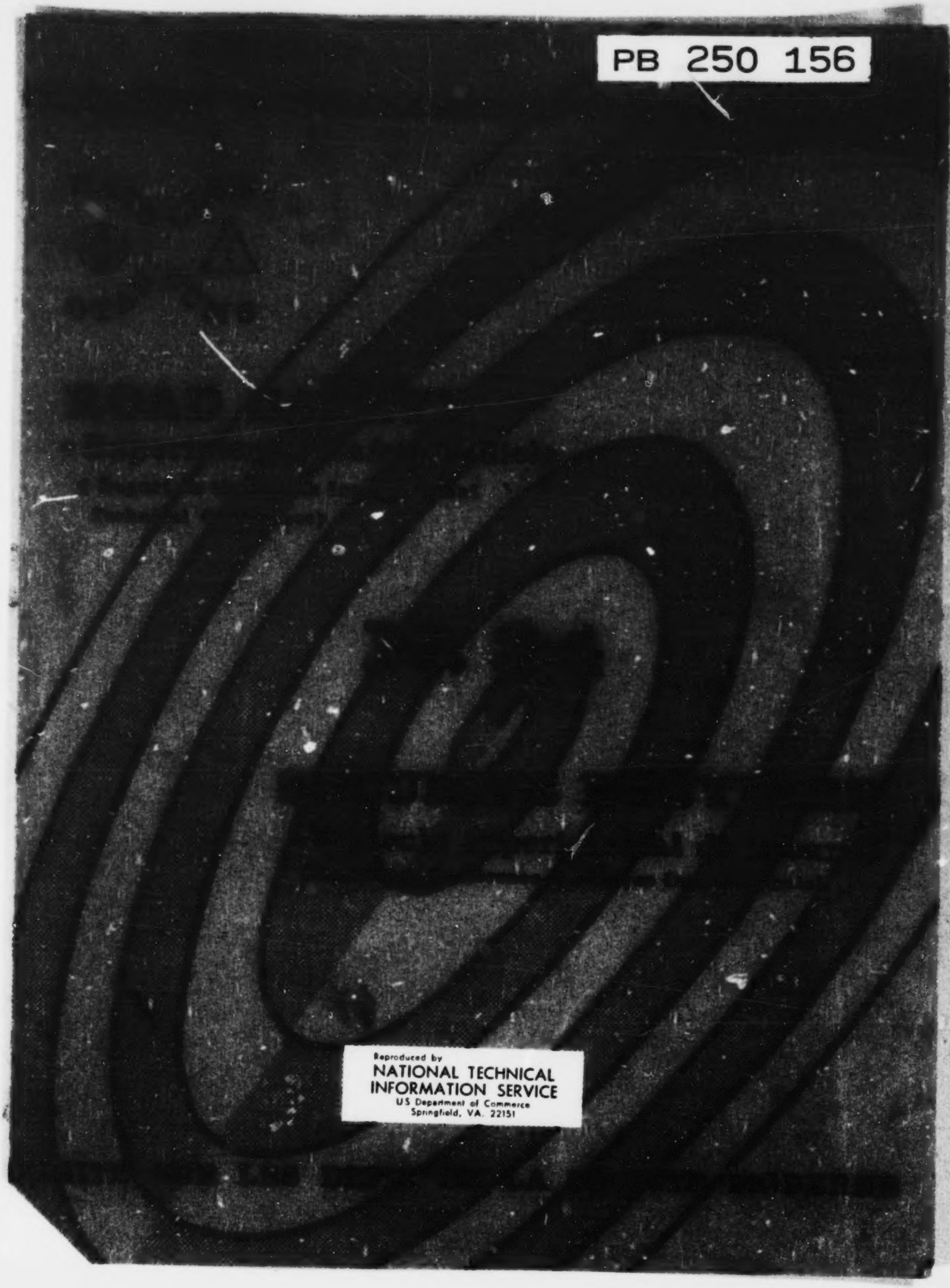

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REPORT ON THE FOURTH INTERNATIONAL TECHNICAL CONFERENCE ON EXPERIMENTAL SAFETY VEHICLES

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Japan Automobile Research Institute

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OTAN CDSM

U.S. DEPARTMENT
OF TRANSPORTATION
National Highway Traffic
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FOREWORD

FOURTH INTERNATIONAL TECHNICAL CONFERENCE ON EXPERIMENTAL SAFETY VEHICLES

This report of the proceedings of the Fourth International Technical Conference on Experimental Safety Vehicles was prepared by the Office of Experimental Safety Vehicle Programs, National Highway Traffic Safety Administration, United States Department of Transportation.

The report includes the conference opening remarks, status reports by governmental representatives and the formal technical presentations by the automotive industries participating. The conference ended with the technical papers presented and the discussions held during the parallel seminars on "Crashworthiness" and "Accident Avoidance," a discussion period on rulemaking and experimental safety vehicles, and summations and concluding remarks by the United States and Japan.

For clarity and because of some translation difficulties, a certain amount of editing was necessary. Apologies are therefore offered where the transcription is not exact.

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INTRODUCTORY STATEMENTS

SECTION ONE

PART 1

DR. GENE G. MANNELLA, *Associate Administrator for Research and Development*
Research Institute
National Highway Traffic Safety Administration

It is my pleasant task to declare this Fourth International Technical Conference on Experimental Safety Vehicles in session.

Let me take this opportunity to tell you that I look forward to serving as your Chairman for the next four days. I am certain that you also look forward to a productive interchange of ideas on automotive and traffic safety.

DR. NOBUTO OHTA, *President*
Agency of Industrial Science and Technology
Ministry of International Trade and Industry

On behalf of the Japanese Government, it gives me great pleasure to welcome you to the Fourth International Technical Conference on Experimental Safety Vehicles. Although Minister Nakasone of the Ministry of International Trade and Industry had expected to address this meeting in person, Diet debates on the budget have prevented him from being here. He has asked me to offer his apologies and to extend his greetings to all of you.

We are honored to be able to host this international technical conference in the historic city of Kyoto. With your permission, I will digress momentarily and touch on Kyoto's historic past. It was toward the end of the eighth century, some twelve hundred years ago, during the reign of the Emperor Kanmu, that the capital, which was then located at Nara, was transferred to Kyoto. This new capital became the focal point for the politics and culture of the time. It flourished and eventually resulted in the creation of the Heian era, which was the most colorful era in the history of the country. Succeeding generations have continued to respect and revere this city and today, it ranks as one of the world's historical legacies . . . .

Three successful ESV International Conferences have already been held in Europe and the United States. At this time, I wish to extend special thanks to the U.S. Department of Transportation for its initiation and sponsorship of these conferences and for its splendid achievements in this area. At the same time, I wish to thank the governments of participating nations and the automobile companies, both of which have accepted the American invitation to join in this undertaking. With the expenditure of an enormous amount of money and time, the participants have built a number of experimental safety vehicles which are on display at Exhibition Hall. Please accept my congratulations on these, the products of your efforts.

Japan was assigned the task of building light passenger cars in the 2,000 pound class. The process of constructing these vehicles has resulted in the accumulation of a vast amount of experience. The problems involved in applying this experience to actual manufacture have yet to be determined. I trust that this conference will be able to determine satisfactory ways in which such problems may be overcome.

Another issue with which we are all concerned is traffic accidents. In 1972, traffic accidents in Japan reached a record high of 659,000. Of these, 15,900 resulted in fatalities. If we were to add the traffic casualties of other countries to this the total number of accidents would be tremendous. Thus, the vehicles, which were once the source of convenience and pleasure, now force us to face a more tragic reality. This situation must be corrected and indeed presents a great challenge to us all. I feel that the solutions for a large part of these problems are dependent upon your wisdom and effort.

JAMES M. BEGGS, *Special Assistant to the Secretary*
Office of the Secretary of Transportation

It is indeed a pleasure to have the opportunity to speak to such a distinguished international gathering, and participate in the opening of the Fourth International Technical Conference on Experimental Safety Vehicles.

Secretary of Transportation Claude S. Brinegar asked me to extend his deep regrets for being unable to personally address all of you here but, I am sure

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you will understand, having just assumed his office there are many urgent matters which require his immediate attention and preclude his attendance here today. However, he is most hopeful of being able to attend the fifth conference in England, and, in any event, I can assure you that both President Nixon and Secretary Brinegar are most gratified by the deep interest and commitment reflected by your participation in this conference.

There is no doubt that the information and ideas that will be exchanged here in the next few days will enhance the motor vehicle safety efforts in each of our countries. Of course, we cannot expect that our discussions will solve all of our automobile safety problems. This conference must be considered simply as one more step in the evolutionary, cooperative development of automotive safety on an international scale. The task we face is indeed immense, but the forthright manner in which we have taken the steps that bring us here today should give all of us confidence in our joint ability to reach our objective safer automobiles. In fact, I feel that our preceding efforts are of such commendable significance that I would like to review them with you for a moment.

I am sure you recall that shortly after taking office in 1969, President Nixon proposed that the scope of NATO activities be expanded beyond military and political considerations to include a consideration of the many problems our industrialized societies now face. The North Atlantic Council adopted President Nixon's proposal and created the Committee on the Challenges of Modern Society, or CCMS, in November of 1969. The CCMS was given the task of bringing increased attention and action to the whole range of new social and environmental problems engendered by the increasingly complex, highly industrialized societies that each of us live in. Toward this end, the CCMS adopted a somewhat unique approach in which a single nation or "pilot" country would assume the primary responsibility for a given area of activity. Under this arrangement the United States became the pilot country for a major effort to reduce the tragic toll, in lives and resources, that each of our countries experience due to accidents on our road and highway systems.

The concept of Government sponsorship of the development of experimental vehicles with designs predicated on safety performance was part of the landmark vehicle and highway safety legislation enacted in 1966 by the United States Congress. In February of 1970, at the first technical meeting on the CCMS Pilot Study on Road Safety, we proposed that the concept be broadened to one of international cooperation. Although NATO, through the CCMS, was used to create a climate and a stimulus which would encourage international cooperation in the area of automotive safety, there was never any intent to limit participation to NATO countries. It was heartening to note how quickly interest and enthusiasm spread. Japan, which is graciously serving as our host country for this conference, was one of the first countries to sign a *Memorandum of Understanding* with the United States Department of Transportation, for the exchange of ESV research information. We have now signed *Memoranda of Understanding* with every major automobile-producing nation in the free world. Safety vehicle designs spanning every important weight class are now being developed. In fact, it is estimated that the worldwide automotive industry is investing 75 to 100 million dollars on ESV development. Three very successful international conferences have been held to provide forums for the exchange of technical information and it is gratifying to see these conferences become more and more comprehensive and productive. The reports that have been published on the international conferences have been complimented by many sources as the most complete in-depth volumes available in the world on vehicle safety. All critical indicators reflect the interest and commitment that each of us has brought to this endeavor.

In closing, I feel I should briefly mention the ultimate goal of this entire international undertaking the translation of ESV performance characteristics into mass-produced vehicles. As we solve more and more of our technical problems, and, in the process, define feasibly attainable performance characteristics, we must ask ourselves how we can put vehicles with the most cost-effective feature of our ESVs onto our nation's highways. This task may very well prove to be more formidable than the development of ESVs. We in the United States Department of Transportation are continually addressing this problem and we are certain that the exchange of technical information and ideas that will take place here in Kyoto will bring us all closer to our mutual goal of greatly improved road safety.

Again, my thanks for affording me the opportunity to initiate what I am sure will be a most productive meeting.

EIJI TOYODA, ***President***

Japan Automobile Manufacturers Association

It is an honor for me, as President of the Japan Automobile Manufacturers Association, to greet you on this occasion on behalf of the automotive industry of Japan. We extend a hearty welcome and wish to express our gratitude to the governments and industries of the participating nations for making this gathering possible. At the same time, I wish to extend special thanks to the U.S. Department of Transportation, to officers of the Ministry of International Trade and Industry of Japan and to other persons involved in deciding to hold this conference in Japan.

Our industry's primary manufactured product, the automobile, now presents us with two major problems: the need to control exhaust emission and the need to ensure greater safety. These are both issues of importance, whose early solution must be effected on the international level. This is a task for which we must all be responsible in order to promote the further happiness of mankind by realizing a "Harmony between man and vehicles" and a "Harmony between society and vehicles."

Automobiles are now an international commodity because they have features which permit them to be utilized by people of all nations. This common use, surpassing national boundaries, necessitaties an international approach to dealing with these two problems.

For the next four days, this Fourth ESV International Conference will function as a realization of this approach, with participation by representatives of governments and private industry. It is most significant that the goal of automotive safety is to be discussed in an atmosphere of international cooperation. I have complete confidence that the presentations of the ESV program status reports to be given by the participating countries and the deliberations on future safety standards will have fruitful consequences.

It is my sincere hope that the results of the research and development which will be presented at this conference will be incorporated into a finished vehicle which will meet the safety requirements demanded by legislation. It is also my hope that such legislation will be universally recognized and accepted.

As is the case with all of you, we of the Japan automobile industry are directing our best efforts toward the fulfillment of the ESV program. This conference will give us all an excellent opportunity to exchange ESV technical information and views with representatives of participating countries. Let us all move forward together for the development of safe vehicles for the achievement of a "Harmony between man and vehicles."

DOUGLAS W. TOMS, *Administrator*
National Highway Traffic Safety Administration

On behalf of the National Highway Traffic Safety Administration, I am very pleased to welcome you all. Many of you have traveled considerable distances to participate in this conference, and we are very pleased to welcome you here.

In particular, I would like to welcome all those participating governments. As many of you know, much of the work of the ESV revolves around international and United States rulemaking. So this is a special opportunity for us all to visit with our government counterparts, and also for those of you in industry to get together and work with these government regulators.

I especially want to issue a special welcome to all of the factories and to all the automobile manufacturers. After all, it is you folks who build the cars, and who are working so hard to develop these ESVs.

I also would like to pay special recognition to the many people who have worked so hard to put this conference together. As we look back at the first, second, and third conferences, I think we are all aware of how much work goes into this kind of an effort. So a very special thanks to our colleagues in the Japan Ministry of Transportation and the Japan Ministry of International Trade and Industry for their particular efforts in making the arrangements for the conference.

I would also like to pay special tribute to the Japan Automobile Manufacturers Association and to the Japanese auto companies who have taken on the burden of making the arrangements for cars, the materials, the microphones, the interpreters, and all the other things that are so difficult in putting on a conference. So to JAMA for helping us all in getting here, and for assisting MOT and MITI in all the details, a special thanks to you.

I would also like to make a comment to our own people — our thanks to Gene Mannella and Pete Shively and so many of our ESV staff who I personally know have been working day in and day out for the past six months to make this sort of thing happen right.

Having been to all of the ESV conferences, I must confess that I was very impressed at Stuttgart and I think that if we are really talking about one-upsmanship, this is a magnificent facility. So to all the people here in Japan who worked so hard to put together such a conference I must express my awe and my admiration. This conference center is probably one of the finest facilities that I have ever seen.

So, again, a special thanks for making this facility available, the outstanding ESV exhibit room, the software rooms, and all the other facilities that we will be using during this week.

I personally look forward to a great conference. I feel that there has been a maturation process. As I think back to our earlier efforts, we were at least in some part feeling our way. I thought last year we brought a lot of it together. It is my hope that this fourth conference will give us an opportunity to really relate the experimental safety vehicle work to our rulemaking and therefore give us the opportunity to exchange the kind of information that will help us save lives and reduce injuries on our world's highways. Thank you.

STATUS REPORTS BY GOVERNMENT REPRESENTATIVES ON THEIR EXPERIMENTAL SAFETY VEHICLE PROGRAMS

SECTION ONE PART 2

UNITED STATES

JAMES M. BEGGS, *Special Assistant to the Secretary*
Office of the Secretary of Transportation

At this time it is my distinct pleasure to present the status of the United States Experimental Safety Vehicle (ESV) Program and its relationship to the "Program Plan for Motor Vehicle Safety Standards."

As you will recall during the Washington Conference last May, we summarized the results of the AMF and Fairchild "Prototype ESV Test Program" and drew some tentative conclusions as a result of those tests. We also discussed a proposed schedule for the follow-on "ESV Optimization Program," which I will discuss in more detail a little later. Further, both the General Motors Corporation and the Ford Motor Company presented the results of their excellent ESV programs. Finally, we agreed to indicate the relationship between the ESV Program and the "Program Plan for Motor Vehicle Safety Standards."

Since the Washington Conference, we have announced the selection of the AMF Prototype ESV as the winner of the competitive test phase. AMF is now conducting a tradeoff and integration study. Upon receipt of the study results, the Department will determine whether or not it is cost effective to proceed with the fabrication of optimized prototypes. During the United States' technical presentation on Thursday morning, you will receive a comprehensive review of AMF's progress.

On February 13, 1973, in Washington, D.C., the Secretary of Transportation officially accepted the Fiat "500" ESV. This acceptance resulted from negotiations with the Italian Government and the Fiat Motor Company for the exchange of Fiat ESV's for test purposes. These tests will address the large car/small car crash problem and will add to our understanding of the problem of vehicle aggressiveness. The initial results of our evaluation of the crash tests between the large AMF ESV and the small Fiat ESV will also be given during the U.S. technical presentations on Thursday morning. I should like to take this opportunity to express to the Italian Government and the Fiat Motor Company the deep appreciation of the United States for this significant contribution to international cooperation in the field of automotive safety.

In regard to the "Motor Vehicle Program Plan," our current thinking on a revised rulemaking schedule calls for the issuance of a system-performance standard which will increase the required fixed rigid barrier impact velocity from 30 mph to 40 mph in August 1979. In reaching this decision, consideration has been given to (1) the information concerning feasibility and lead time furnished by the international and domestic manufacturers; (2) the results of NHTSA research on structures and restraining compatibility apart from the ESV; (3) the insight gained through the ESV program, and (4) best estimates of the types of collisions that are causing the most serious injuries and fatalities. These estimates are based on NHTSA, as well as other, accident investigations and statistical data.

Discussing these points in inverse order, a picture is emerging from accident statistics that indicates that the so-called "barrier equivalent speed" is lower than previously thought for many car-to-car collisions. This is particularly true for side impacts, whose contribution to the total problem had been previously underestimated. With regard to barrier equivalent speed, this problem still awaits a quantitative solution, but NHTSA is actively investigating variable yielding barriers, both fixed and moving, in an attempt to obtain test requirements which are more representative of actual accidents. This work has not progressed to the point that we can predict whether these developments will affect the requirements for the planned standards revision in 1979 or 1980. However, it is believed, from mutual discussions, as well as from NHTSA calculations, that this is a more meaningful approach if the requirements are properly established.

The ESV program has established the technical feasibility of producing structures that are capable of 50 mph fixed rigid barrier impacts, yet have low aggressiveness in impacts with small cars and/or more compliant vehicles and/or at low and intermediate barrier equivalent velocities. Although these vehicles were over-designed and very heavy, there is every reason to believe that much of the excess weight can be eliminated in another design iteration.

NHTSA contract research programs are actively pursuing a better understanding of the structure and restraint compatability problem. This work is based on expanding earlier work presented by ESV manufacturers at earlier conferences. Briefly, the combined crush-interior distances required for reasonable restraint forces are large in high speed impacts. This is particularly true if there is a lag in restraint application while occupant on-set rates are kept at, or below, previously recommended values. For this reason, also, it appears that the published schedule could be stretched out, although it is known that 40 mph performance is feasible under carefully controlled conditions today.

With regard to lead time and other manufacturing considerations, the manufacturers are all aware of the problems which are involved. It can be stated that they vary from manufacturer to manufacturer and that rulemaking can probably never accommodate the entire spectrum if it fulfills its responsibility to the United States' public. However, these problems will be considered to the extent possible within the responsibilities and constraints that exist.

The 40 mph rule will be supported by the follow-on, optimized, ESV which will target for higher performance. In parallel, exploration of the upper limits of practical feasibility will be pursued by the new, longer-range research safety vehicle project. This study will build upon the foundation of data that exists in the international ESV program and investigate the tradeoffs and technology required to reduce vehicle weight while maintaining the 50 mph goal.

I would like to take this opportunity to clarify the schedule and plan of the U.S. ESV program. We intend to continue the "AMF Tradeoff and Integration Study." This could lead to the development, fabrication and testing of two optimized ESVs. Our present thinking is a car capable of attaining barrier speeds of 45 mph. These requirements support the rulemaking program plan which now reflects a 40 mph requirement in the late 1979 time frame.

The "Advanced State-of-the-Art Research Safety Vehicle Project" is entering the conceptual and feasibility study phase. In mid-February of this year, we forwarded copies of our initial performance specification for your review and comment. I must reiterate that these initial requirements are tentative goals that must be refined as the project progresses. In summary, through this project we intend to push current technology to develop imaginative safety designs that can have practical application in the next decade. We are considering weight spans of from 2,500 to 3,500 pounds with barrier speeds hopefully at 50 mph. I anticipate that during the discussions scheduled for Friday we will have the opportunity to discuss in more detail your reactions to these points.

In keeping with our policy of addressing the safety needs of the total vehicle transportation system, we initiated this year the "Experimental Safety Motorcycle (ESM) Project" which has as its ultimate objective of this development a systems safety performance specification for this class of motor vehicle. Other classes of vehicles will receive attention under our special-purpose project which we plan to initiate next year. The safety school bus is a priority effort within the special purpose project.

As the international ESV program, initiated under the aegis of the NATO Committee on the Challenges of Modern Society, moves from the development phase to the testing phase, I again extend to each of you the invitation to send ESVs to us for test purposes. I hope that the exchange of the Fiat ESVs signifies only the beginning of a major international ESV test program.

STATUS REPORTS BY GOVERNMENT REPRESENTATIVES ON THEIR EXPERIMENTAL SAFETY VEHICLE PROGRAMS

SECTION ONE

PART 2

JAPAN

YASUO NAKAMURA, *Director*
Automobile Division
Ministry of International Trade and Industry

I am pleased to give this progress report on the development of ESV in Japan.

In November 1970, in response to the request of the U.S. Department of Transportation, the Japanese Government signed an agreement on the joint development of ESV. On the basis of this agreement, and in cooperation with the Japan Automobile Research Institute, specifications for ESV cars in the 2,000 pound class were prepared. The government then asked automobile manufacturers to participate in the development of ESV cars based on these specifications. Three manufacturers, Toyota Motor Company, Ltd., Nissan Motor Company, Ltd., and Honda Motor Company, Ltd. were authorized to undertake this project. Toyota was assigned the task of building a vehicle of 900 kg (2,000 lbs.) for two occupants; Nissan, a 1,150 kg (2,500 lbs.) car for four occupants, and Honda, a 750 kg. (1,050 lbs.) car for four occupants. In order to finance the building of prototype vehicles, the Ministry of International Trade and Industry granted 596 million yen to Toyota and Nissan. The Japan Automobile Research Institute received a 704 million-yen grant to establish facilities for the general testing of vehicles. Delegates from each company will report on the development of these cars this afternoon. At this time, however, I will briefly outline the progress made by each manufacturer.

As I mentioned earlier, Toyota was assigned to build a 900 kg. car for two occupants. By the end of September 1970, the company had built twenty initial prototype cars. These were tested for accident avoidance characteristics and crashworthiness. As a result of these tests, this April (1973) Toyota will begin to build the final test vehicles. Ten of them are to be delivered by September of this year as prototypes for official experiments. The weight of the car will be increased by about 20 percent.

Nissan Motor Company was asked to develop a 1,150 kg. car for four occupants. For the first stage, 120 already existing vehicles were converted for experimental purposes. A wide range of data has been gathered. This includes data on accident avoidance and crashworthiness, as well as data concerning the occupants' protection system. In September 1972, Nissan began the second phase of prototype production. During this period, 38 vehicles were subjected to repeated experiments on the energy absorbing features of individual components and the functioning of passenger protection devices. Results of these tests indicated that under crash conditions, the vehicle was likely to remain intact. However, the outlook for occupants was not considered as satisfactory. Emphasis is now being placed on improving the chances for passenger survival. Production for the final prototypes for official experimentation will begin in July of this year. In September, ten cars are expected to be ready for delivery. The weight of these cars will be increased to 1,200 kg.

Honda Motor Company undertook the building of a 750 kg. ESV car for four occupants. Honda adopted a system according to which parts would be developed and the vehicle would be assembled upon the satisfactory completion of the various parts. This car is expected to weigh over 800 kg. and is aimed at raising the general standards for accident avoidance capability, as well as increasing the possibilities for passenger survival.

By September 1973, Toyota and Nissan are expected to have ten ESV cars completed. Honda will complete its vehicles by the end of 1974. At that time, the Japan Automobile Research Institute will do the official experiments on these cars.

In order to do the necessary research, the Institute created an ESV Committee with four subcommittees. These groups have been conducting research and engaging in experimentation on visibility, maneuverability, braking characteristics and crush tests. As I mentioned earlier, a 700 million-yen subsidy has been granted by MITI (Ministry of International Trade and Industry) to establish complete facilities for the testing of vehicles. I am confident that given the staff and the experimental procedures that are now being used and will be used in the future, the Japan

Automobile Research Institute may be rated as one of the finest experimental institutes in the world.

The members of this meeting are to discuss all aspects of ESV. These discussions will be based upon actual research. Your efforts, I am sure, will greatly influence the world automotive industry. In closing, let me say that the Japanese Government, which feels that Japan is making favorable progress on ESV, is most interested in these deliberations in Kyoto.

STATUS REPORTS BY GOVERNMENT REPRESENTATIVES ON THEIR EXPERIMENTAL SAFETY VEHICLE PROGRAMS

SECTION ONE

PART 2

GERMANY

HELMUT WAGNER, *Deputy Assistant Secretary*
Federal Ministry of Transport

Speaking for the Federal Republic of Germany I am honored to present a brief report on the present status in the work related to experimental safety vehicles.

During previous international safety conferences you have received information on experimental safety vehicles developed in my country, according to the original ESV specifications issued in 1970. The tests related to these vehicles have been completed.

The research, however, has been continued as follows:

Daimler-Benz has developed and built a new research vehicle, the ESF 22, as a first step in investigating the physical and economical vehicle design possibilities throughout the large range between the technology of today and the original aims of the ESV program. Also, this vehicle — as a unit — does not represent an economically feasible solution for a series-production passenger car.

Singular results of this research, however, will be evaluated in the traditional manner for their eventual application in the continuous development of future series-production passenger cars.

Volkswagenwerk AG has finished the tests according to the German and U.S. ESV specifications. The main requirements, specifically in the area of occupant crash protection, could be met. A detailed report will be submitted during this conference. As a result of this research work, Volkswagen found that in several important areas there were not sufficient data available for a final judgement on the benefit of this ESV car. Therefore, it was decided to expand the scope of work to the following specific areas: aggressiveness, biomechanics, handling and stability and benefit cost analysis.

BMW, Opel and Porsche have continued their work on vehicle subsystems. The results of these activities will be presented during this conference. The experimental safety program of BMW is continually based on studies and tests to maintain good accident avoidance characteristics on vehicles equipped with advanced occupant protection devices. They are attempting to provide more feedback information to the driver through the steering wheel and seat, and by suitable instrumentation, more information on distance from the other objects, and required stopping distance and accelerations.

Opel continues to conduct a component test and development program targeted to develop optimized solutions which can be incorporated into future prototype vehicles. In both areas — occupant protection and accident avoidance, the measures will be selected based on actual traffic accident data and the evaluation of benefit/cost calculations.

Porsche continues to improve primarily the safety of production cars, especially in the handling areas. But Porsche agrees, that the passive safety should not be neglected.

The Association of Liability, Accident and Automobile Insurers HUK — Verband) continues their efforts in evaluation road traffic accidents. HUK has investigated the merits of anti-lock systems and has attempted to find a relationship between collision speeds in actual accidents and crash test velocities.

The members of the German delegation appreciate this opportunity to again exchange ideas on safety improvements and will attempt during the following months to evaluate the results of the conference.

STATUS REPORTS BY GOVERNMENT REPRESENTATIVES ON THEIR EXPERIMENTAL SAFETY VEHICLE PROGRAMS

SECTION ONE PART 2

THE UNITED KINGDOM

H. TAYLOR, *Head*
Safety Department
Transport and Road Safety Research Laboratory

In Britain we elected to begin our programme by adopting a safety-systems approach, believing that this would provide more flexibility, initially, for establishing improved, safety-performance requirements for complete vehicles.

Our starting point was the study of road accidents and their casuation and consequences. It soon emerged that boundaries could be defined, for example of impact speed, which would encompass the majority of real-life accidents and that the problem was not simply one of protecting vehicle occupants in an individual vehicle.

It was gratifying to see these points recognized at the Third ESV Conference and I hope we can make further progress on this occasion with defining the boundaries and dealing with the problems of compatability between different classes, whether of large and small cars, or of cars and pedestrians.

CONTRACTS

During the past year we have proceeded with the contracts previously notified and since the last meeting further contracts have been placed. An up-to-date list of these contracts is given in the UK papers for this meeting; the total value of contracts placed is approaching £1 million. Discussions are still proceeding on the large-scale trial of current production cars with non-locking brakes.

A large part of the international programme is concerned with occupant protection, but accident avoidance is the primary goal. Non-locking brakes are one of very few prospects for making a major advance in accident avoidance; however it remains to be established whether the resulting benefit in braking and handling will be offset by adverse driver behaviour, and it would be a major feature of any trial to investigate this aspect.

RESEARCH ON SAFETY STANDARDS

As a result of our work to date on accident avoidance, occupant protection and pedestrian safety, we feel able to begin defining realistic vehicle safety performance requirements of the future, for complete vehicles. An important part of our technical presentation will be concerned with the requirements chosen, which we believe will have the maximum beneficial effect on the accidents which occur on our roads, without raising prohibitive vehicle weight/cost problems. These requirements should not prevent the continuing production of cars which are fully acceptable to the public and our exhibit at the Third ESV Conference illustrated some of the possibilities.

Representation of the human body for testing purposes continues to be an important part of the programme. A contract has been placed to develop a test dummy with the following essential features; its performance must be related to the results and criteria derived from real-life accidents; it must yield reproducible results and be relatively insensitive to the setting-up procedures employed in testing; the initial cost must be modest and the dummy should be re-usable at minimum repair cost (see page 247 for prototype dummy).

In formulating requirements, on which much work remains to be done, we are of course prepared to use the results of international studies on accident data, such as the CCMS project, in addition to UK statistics.

EXPERIMENTAL SAFETY VEHICLES

The British car safety programme has always contained provision in Phase 2 for the construction of prototypes incorporating various groups of safety features. We consider that this stage has now been reached and during the next year several complete cars will be commissioned, covering a range of vehicle sizes. These will proceed in parallel with research on safety standards to demonstrate not only that higher safety requirements can be met technically but also to establish, where possible, the tradeoff between safety performance and cost. The compatability of different sizes of cars will be studied and a start will be made on embodying the results of research into pedestrian safety features.

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IMPLEMENTATION OF RESEARCH

During the Third ESV Conference reference was made in the *UK Status Report* to the need for strenuous efforts to resolve the differences between vehicle regulations worldwide. The UK team looks forward to progress here in the seminars and in the planned discussion of safety standards on Friday.

EXCHANGE OF INFORMATION

We shall in the future be issuing progress reports at regular intervals which will of course be freely available to countries participating in the ESV programme. The first of these is available now and the progress reports will be augmented by detailed contract reports as the work proceeds.

STATUS REPORTS BY GOVERNMENT REPRESENTATIVES ON THEIR EXPERIMENTAL SAFETY VEHICLE PROGRAMS

SECTION ONE

PART 2

ITALY

PROFESSOR AUGUSTO SIRIGNANO,
Director General For Motorization
Ministry Of Transport

When I received the invitation from the United States Department of Transportation (and from our Japanese colleagues) to take part in this fourth conference on the safety vehicles, I had on my table statistics on the road accidents recorded in Italy in the course of 1972. An impressive number of deaths — almost 10,000 — of serious or less serious injuries — over 240,000 — of families destroyed, of activities disrupted and of property irremediably lost.

I believe I am not mistaken in stating that so heavy a toll of suffering and bereavements cannot leave indifferent all those who consider it a civic duty to undertake more effective commitments to ensure that this tragic trend is effectively fought, or at least contained within limits as restricted as possible. The vast and selected participation in this conference is a visible and symptomatic testimony of this intent.

The Italian delegation has welcomed with special pleasure the opportunity offered it to get to know the latest efforts being made in the various countries concerned with the sector of motor car safety, the progress which, despite difficulties and considerable costs, the industries concerned have managed to achieve and the programmes which they intend to work out.

This type of meeting, like every other highly qualified gathering, generally has the merit of speeding up the framing and development of the chosen plans, since everyone is led to conclude in concrete manner the cycle of their own experiences and to note in a wider and more open discussion the validity of the results achieved.

However, it is essential that this quite legitimate concern not be transformed into a sterile competition, because that would lead to a certain prejudice of the objective and rational evaluation of the requirements, and thus a negative repercussion on the choices and final orientations to be adopted.

Furthermore, in order to maintain a high technical level in this type of meeting it is necessary to be in a position to present the results of complete cycles of studies.

In the very short space of time — only nine months — which have passed since the Washington Conference, the progress made in carrying out the study programme we had proposed for ourselves together with our other European partners has necessarily been limited. In this connection, in order to ensure a more ordered and constructive development of our programmes, I will take the liberty of drawing to your attention the advisability of leaving longer intervals between the dates planned for our meetings.

Nevertheless, the short time which has passed has served to frame, in terms more closely adhering to our reality, the problems of realizing a safe experimental vehicle; of opening up wider perspectives to include consideration of all components of traffic, in order to verify certain important working hypotheses on which to base our action; and finally to lay the bases to acquire new and more effective forms of intervention at every level, including that of governments.

As regards the plan for a safe vehicle, we have consolidated the previous conviction that all vehicle traffic should be made compatible — within reasonable weight limits.

The Italian Government has viewed with particular interest the delivery, last February to the U.S. Department of Transportation, of the Fiat 1,500 pound prototype, intended to undergo crash tests at high speed with an ESV prototype of the 4,000 pound class made in the United States. The highly significant purpose of such tests is to verify, experimentally, the behavioral characteristics of two widely differing ESV prototypes, as regards both weight and planning.

If it should prove impossible to achieve a compatible traffic mix of vehicles with similar and substantially different weights, it will be necessary to utilize resources (at present not available) to build impressive road infrastructures or, alternatively, to exclude motor cars not having a limited weight.

The dilemma was explained at the Washington

Conference, and active efforts are being made to work out satisfactory solutions. So it is easy to understand the desire for this conference to produce clear indications of new orientations to be followed in order to reach speedily, and by the least difficult and shortest road, the objective chosen in the general interest, and for which we are all gathered here today.

First of all, we feel that specifications should be laid down for the ESVs which will be correlated to the situation of the circulating vehicle in Europe and markedly, as far as we are concerned, to the particular conditions of capacity of the majority of Italian users. But, outside of and beyond any strictly sectorial consideration, perplexities still exist at the technical level also for the overall acceptance of the norms fixed by NHTSA for the realisation of prototypes of Experimental Safety Vehicles.

The only goal of the whole programme should be the safety of occupants. Accordingly, certain structural prescriptions are useless (which it is always possible to satisfy if wished, whatever their gravity) which are not indispensable to reducing the consequences of a crash on the human body to a level bearable by the body itself; this is the departure point of the whole programme.

In September last year we submitted to NHTSA a list of tests to be undergone by the 4,000 pound prototype ESVs built in the United States, which in our opinion seemed partly superfluous if not in fact conflicting with one another.

A few days prior to our departure for Japan we received the proposed specifications for an ESV of 3,000 pounds which will be discussed and drawn up in its final form at the end of this conference. Unfortunately, we did not have the time required to examine adequately this documentation and, accordingly, we are not in a position to form a sufficiently grounded judgment in this regard, nor, consequently, to appraise what proportion of the remarks earlier forwarded to the NHTSA have been included in the new specifications.

Now let me make it clear that these considerations do not signify any reserve regarding the efforts which the United States is carrying out in the field of road traffic, and whose responsible commitment we all approve, but only to express by way of constructive criticism, at the same time indispensable, the conviction of drawing closer to positions divergent in certain respects for the purpose of better defining the terms of a more coordinated and thus more fruitful collaboration.

We are of the opinion that our Japanese colleagues also are applying themselves in this direction on these bases, so that it may seem superfluous to call on them to ensure that, to such end, the bases are laid for a closer and more stimulating mutual collaboration.

In fact, the organisation of this conference, which calls for considerable accommodation facilities, indicates the exact extent of the interest that this marvelous country pays to facing up to so complex a problem and one of such vital importance.

As far as we specifically are concerned, I would like to stress that the Italian motor car industry has carried forward considerably the programme assigned to it. Fiat has devoted itself particularly to the development of passive safety, while Alfa Romeo has undertaken a perfecting of studies on active safety.

The three prototypes exhibited by Fiat and the results achieved by the other industries and institutions represent in a certain sense the tangible expression of the efforts made and the most valid testimony of the seriousness with which commitments are honored.

I think we can say with a clear conscience that all that Italy could do, within the limits of its own resources, conditioned as it was by rigid priority choices, has undoubtedly been done. However, we are nonetheless aware that, despite our most consistent efforts, we still cannot see even in the distant future a time when a safe vehicle will be placed on our roads.

Accordingly, we consider it necessary to include the plan for the safe vehicle in a wider framework; in other words, we feel sure that it is essential to speed up the systematic and coordinated study of the various factors of safety, namely the man, the road, and the vehicle.

The results achieved in such research can determine positive influences on the planning of safe vehicles which, in regard to their planning and realization, should constitute a happy synthesis of the experiences gained. To this end, a decisive contribution could be offered by a speedy conclusion of the work entrusted to the Special Committee of CCMS, which is charged with the multi-disciplinary inquiry into road accidents.

The Italian delegation cannot refrain from pointing out how vain it is to set final objectives in the field of safety if we do not first arrive at the definition of the characteristics of an anthropomorphic dummy which corresponds, as far as possible, to real people so that we can guarantee the indispensable repeatability of experimental results.

In addition, a specially detailed examination should be directed to the inquiry on the limits of survival of the human body in objective terms. Basically, we are convinced that the programme for the safe vehicle should include other important aspects which are not exclusively referrable to the vehicle, such as biomechanics.

It is obvious that such a result will be more easily achieved if we continue the cultivated and, better still, integrated sense of friendly cooperation which has so far marked our research activity.

Last but not least, as a further confirmation of the concrete and present interest which the Ministry of Transport of my country pays to the problem of safety, convinced as we are that human life is the most precious gift of creation, I wish to inform you that we have now reached the advanced stage of a highly specialized Experimental Center to be set up at Rome on which to confer in this specific field the tasks of the utmost importance as regards study and experimentation.

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FRANCE

MICHEL FRYBOURG, *Director*
Institute of Transportation Research

Two years ago, the First International Technical Conference on Experimental Safety Vehicles was held in Paris. It is now possible to make a first estimate of the effort accomplished. The pilot study on road safety undertaken by the United States at the request of the CCMS was based on eight projects — three of them dealing with the protection of passengers — passive restraint devices, accident investigations, and experimental vehicle realization.

It is most important to clearly point out the connection between these three projects. The one on passive restraint devices is to be considered as short-dated, the other project, considered as long-dated, concerns experimental vehicle realization which allows the estimation of the technological limits. The third project deals with accident analysis: it must allow the restatement of an analytical method aimed at evaluating the effectiveness of the measures to be taken in order to protect the passengers of the vehicles.

A safety program involves a set of lines of action in which the effort to be accomplished has to be balanced with the effectiveness realized by each of those actions. This ponderation is liable to be reviewed and it is important to be able to manage the program, to "monitor" it, as some say, in light of the first results obtained.

This matter will be further developed in June, during a meeting held in Paris, organized by the United States, the pilot country, before the closure of the pilot operation planned for November.

We do not wish to anticipate the conclusions to be drawn from this wide confrontation of safety policies of different participating countries. However, in limiting the objectives to the protection of passengers, it is important to point out critiques on the three projects adopted by the pilot study.

FIRST REMARK

The safety belt remains the most efficient equipment to assure the protection of passengers, since the passive restraint devices have not as yet satisfied the hopes founded in them.

The use of the safety belt is the most economical and surest way to reduce injuries among passengers. Thus, it is essential to focus the maximum effort on research of safety belts of a more convenient use, as well as on different incitement methods leading all passengers to a systematic use of safety belts.

SECOND REMARK

The technological assessment is as important to develop as the technology itself, for it is the only way that guarantees an optimum assignment of limited resources in order to achieve the pursued objective of stabilization in road accident deaths.

The project concerning accident investigation is less ambitious: simply the restatement of a methodology. The maximum effort was focused on the vehicle technology in order to satisfy the technical specifications whose merits have always been discussed by different delegations.

Among the difficulties in establishing technical specifications, the most fundamental point is the evaluation of the degree of seriousness inflicted on the passengers in true crashes, from which the levels of useful performances required in test crashes have to be drawn.

We have already fixed the objectives of performance for the restraint devices according to an accurate analysis of accidentological observations.

The application of this method shows that a great efficiency in the protection may be obtained from the valid restraint devices for velocity variations of about half of the limit speed values adopted by most countries for their ordinary roads. We believe that a prior action has to be undertaken in that direction. This does not mean that there is nothing to be done on structures, notably to avoid the collapse of the passenger compartment which constitutes a braking space for the passenger, necessary to assure the efficiency of the restraint device. But, this has to be done with discernment, taking into account the compatibility problems caused by the differences of

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weight. One must, notably, avoid to adjust the stiffness to the mass and to systematically focus the dissipation of energy by deformation of the light car.

At a time, when the American Congress has just set up the Office of Technology Assessment (O.T.A.), it seems very useful to insist on the importance of technological assessment.

In France, the themes of our Programmed Thematic Action on vehicle safety are opened. In other words, no limit has been required for the performances to be achieved by experimental subsystems. We intend to continue in this direction through 1973, but it will be in 1974 (if the assessment studies are sufficiently advanced) that quantitative objectives can be adopted. An important part of our financial means will be assigned to these assessment studies.

Our program forecasts:

- The setting up of a photographic library on controlled crashes. The aim of this library will be to allow a possible scaling of true crashes by means of an index of the violence of the crashes. Thus, the statistical distribution of dissipated energy during the crashes may be studied, allowing the estimation of the usefulness in fixing at such-and-such level the test velocity, in order to appreciate the vehicle qualities concerning the protection of the passengers. One has to bring out of the present empirical dicussions on the opportunity of the 50 mph for the head on crash.
- Experimental study of the effectiveness of anti-locking brakes. Anti-lock braking systems are technically interesting but their production cost, even in mass production, will certainly be elevated. It is essential to have enough accurate information concerning their effectiveness, in a true situation of driving, before considering making their use compulsory.
- Study of the feasibility of a black-box. Accurate knowledge of the true conditions in which accidents occurred (velocity, acceleration, wheel-handling) is necessary to select the right preventive measures in accident matter. The setting up, on an important sample of vehicles, of a black-box which records these parameters would be a great step forward in safety.

The struggle between the cannon and the armour cannot, in fact, surpass a reasonable limit. Once we have a better knowledge of both crash impact velocities and the limit possibilities of the automobile technology concerning the protection of passengers, we will obtain a convenient adequacy of the cannon, that is to say of observed velocities and observed weights and a convenient adequacy of the armour, that is to say of possibilities of passive or non-passive restraint systems and of the dissipation of energy by the vehicle structures.

It is clear that this adequacy cannot be obtained in satisfying conditions simply by the repression whose level cannot surpass a tolerance level of public opinion, often misinformed on speed dangers.

It is also clear that a general speed policy is to be defined, based on a right appreciation of the compatibility between vehicle conception, road characteristics, climatic conditions and driver behavior.

The interest of assessment studies is double:

- The first is already emphasized: only the assessment studies are allowed to direct the effort for the technological progress in the most profitable way for the Collectivity. The Brooks report on the politic of Science, prepared by a group placed under the sponsorship of the O.E.C.D. demonstrates this clearly enough.
- The second interest, just as important, is that these studies lend themselves remarkably well to an "International Cooperation Action," since the collected informations can be often generalized. The information designs achieve considerable efficiency and precision when they are extended to several countries.

In order to avoid non-tariff barriers at the exchanges, the conclusions of these assessments must be subject to a wide, international, consensus. The definition of the objectives concerns governments while it is up to industry to find the most appropriate means to satisfy these objectives.

There has been an attempt to stimulate the technological progress by the pilot study of road safety; this stimulation will bear fruit, but it could have been better directed. Each country that is determined to cooperate on road safety improvement, must develop assessment studies in order to clearly define the goals to be achieved through technological progress.

My government sincerely hopes and proposes that the assessment studies program will be achieved in an International context which will considerably increase its efficiency.

SWEDEN

GUSTAV EKBERG, *Chief*
Vehicle Division
National Traffic Safety Agency

The Swedish contribution to the international cooperation on ESV is a research project called, "Steerability During Emergency Braking." This project initiated by the Swedish government and the Saab-Scania and Volvo automobile manufacturers was outlined at the previous conference in Washington. The work is continuing on schedule and a progress report will be given during the conference. You may also recall that an experimental safety car has been designed and built by Volvo. The experiences from tests with this car will also be reported. Furthermore, Saab-Scania and Volvo will present investigations on traffic accidents, the main purpose of which is to elucidate the influence of vehicle design on the causes and results of traffic accidents.

Through the joint international efforts on the ESV project the design of future safety vehicles is now emerging. Nevertheless, much work still remains to be done. The results achieved by the participating countries have to be converted into regulations and recommendations that will serve as lines of direction for the car manufacturers in the future. It also is of great importance that this second phase of the project be dealt with through close cooperation among all the nations interested in increasing road safety and under the auspices of an international organization commonly acknowledged.

Although we all agree that the safety vehicle now under creation will have a considerable positive effect on safety, we must not forget other parts of the rather complicated work on traffic safety. The ESV project has, up to now, mainly been devoted to passenger cars, in particular their crashworthiness and handling characteristics. Cars can and must be improved also in other respects, for instance lighting. I am convinced that great improvements can be achieved if we attack this severe problem on a scientific basis, excluding all preconceived subjective and egoistic opinions. Road safety can also be increased by providing against critical situations on roads, for instance designing heavy commercial vehicles and their trailers so that they maintain their stability and roadholding characteristics under adverse conditions. Let us hope that the international cooperation that has been initiated so successfully by the ESV project can, in the future, be focused also on problems of that kind.

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BELGIUM

JEAN-PAUL DE COSTER, *Director*
Foundation for Study and Research
for Highway Safety

At the two last conferences in Stuttgart and Washington, the Belgium delegate explained why Belgium was interested in the development of ESVs.

If we have no car design department we can, in agreement with the other countries of the European Economic Commission or the Economical Commission for Europe in Geneva, implement the positive results from ESV in future car design regulations.

I am also pleased to be able to address you as project leader for pedestrian safety of the C.C.M.S. Road Safety Pilot Study.

If we look at the specifications on which the present ESVs are built, we see that pedestrians have been neglected. Since the number of pedestrians killed annually is roughly equal to vehicle-occupant deaths, not counting the two-wheelers, we have to admit that the specifications give more emphasis to material losses than to pedestrian protection.

With this situation in mind, the pedestrian safety group has presented a draft resolution to the last C.C.M.S. plenary meeting. We hope it will soon be adopted by the governments.

This draft resolution asks for the same maximum human tolerance characteristics for pedestrian collision at 20 mph as for car occupants in collisions at 50 mph. This relatively low-speed limit has been chosen due to the fact that the problem is difficult and has been subjected to a far less number of studies.

I was personally happy to see that the specifications for the 3,000 pound ESV were written in that spirit.

I will conclude by wishing that if a compromise has to be found between the reduction of material losses and pedestrian safety, human life will be first protected.

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SECTION ONE

PART 2

THE NETHERLANDS

JOHAN G. KUIPERBAK, *Deputy Director*
Ministry of Transport
Road Transport Services

As I have already stated at an earlier conference on this subject our government is not effectively participating in the project of the Experimental Safety Vehicle but is, nevertheless, supporting the effort behind this project.

Our industry, i.e., our automobile industry and our components industry, is highly interested in the developments; which is why representatives of our automobile industry are present here.

Up to now very substantial progress has been made by the participating countries and industries who have joined forces in this project. My government is convinced that the findings, improvements and facts that are already known, will, in the very near future, be used step-by-step in mass-produced vehicles and in that way help all of us to produce safer automobiles and consequently decrease the death and casualty rate on our roads.

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SECTION TWO

PART 1

INTRODUCTION

KIYOSHI YAMOTO, *Chairman*
Air Pollution and Safety Committee
Japan Automobile Manufacturers Association

As the chairman of the Automotive Hazards and Safety Committee of the Japan Automobile Manufacturers Association, Inc., I am most interested in hearing the technical presentations on the ESV program, including those by Toyota Motor Company, Nissan Motor Company and Honda Motor Company.

Toyota, Nissan and Honda have been working under an agreement with the Japanese government which enables them to manufacture trial ESV vehicles and deliver them to the Japan Automobile Research Institute for inspection. After these vehicles have passed inspection, they will be displayed at a public exhibition, which is scheduled to be held at the end of next year. Fortunately, work on these vehicles has progressed quite favorably, and the participants are able to give us a technical assessment of their efforts. However, it remains to be seen whether the features of these prototype vehicles can be successfully incorporated in the production line so that they conform to existing laws and regulations.

I am pleased to be able to say that the rate of traffic accidents in Japan is declining and that the number of fatalities is also decreasing, compared to the number of vehicles in use. In this respect, Japan may be ranked as one of the nations with the lowest number of traffic accidents and fatalities. In addition, Japan has a low vehicle-occupant accident rate. It is our expectation that these satisfactory developments will be supplemented by a special Japanese safety program which is currently being developed and that this program will, in turn, be helpful in the implementation of the ESV project.

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART I

OUTLINE OF TOYOTA ESV PROGRAM

TATSUO HASEGAWA, *Managing Director*
Toyota Motor Company, Ltd.

We in Toyota have completed the first prototypes in accordance with the Japanese ESV specifications, and have compiled various evaluation tests using such vehicles. Based on what we have learned from these experiences, we have completed the design of a prototype for a second series and have embarked on their fabrication. About ten of these vehicles will be delivered to our government. The delivery date will be within this summer, earlier than the end of this year, as originally scheduled.

During the course of this project, we have, in some areas, exceeded the Japanese ESV specifications by utilizing the current automobile design techniques and ordinary materials for mass-production cars. In reviewing the development stage we found that in the area of accident avoidance, particularly in braking, handling and steering, we have reached the desired level. In the area of crashworthiness, however, we have not yet achieved all the requirements. These points are being improved in the design of the second prototype and we expect that all the targets can be achieved in the final stage.

Nevertheless, in view of the overall automobile safety and marketability, there still remain numerous problems. One is the relationship between the complex, real-world accidents and the test methods specified, such as the perpendicular barrier impact or the pole impact; our ESV is designed to meet the very restricted impact requirements of Japanese ESV specifications. Further study is needed on the protection from vehicles with different weights and crashworthiness.

Secondly, there is the problem of out-of-position occupants, as opposed to dummies which are properly seated in the design-seating positions in the tests. There is also the problem of dubious resemblance of the dummies to human bodies, making valid evaluation of human survivability very difficult indeed.

Thirdly, there are certain areas where quantitative evaluation is possible but the relevance to safety is obscure, such as steering characteristics and visibility. Here further study is needed in terms of ergonomics.

Then there is the problem of excess weight and the price that is inevitable in order to satisfy all the requirements. I feel that some revision must be made to the Japanese ESV specifications. In this connection, we need precise accident statistics and the establishment of new effective vehicle safety specifications where superfluous requirements are omitted, thus minimizing the cost to the consumers.

I must also point out that there are phases of safety that automobiles alone cannot achieve. Traffic safety is possible only when human and environmental factors come up to the desired level. It will be only then that practical and marketable safety vehicles will come into reality. We at Toyota are determined to continue efforts toward such marketable safety vehicles.

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THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART 1

BRAKING, HANDLING AND STEERING

TAKASHI BABA, *Assistant Manager*
Proving Ground and Dynamic Laboratory
Toyota Motor Company, Ltd.

It is a great honor for me to present the results of the first tests of the Toyota ESV, with regard to braking, handling and steering. These characteristics are greatly important in accident avoidance.

Accidents are considered to occur when matching between man, vehicle and the environment is broken down. All the braking, handling and steering characteristics work effectively only when well controlled by the driver within the traffic environment. The performance shown as ESV specifications, in this sense, only shows a shallow part of references, and can not extend directly to the sharp reduction of the real-world traffic accidents. Considering such conditions in a wide range of vision, we have propelled various technical approaches to the ESV development.

Figure 1 shows a wind tunnel test with a 1/5 model of Toyota ESV.

Figure 2 shows how we measure a vehicle's moment of inertia with a new type apparatus.

Thus, we have made the basic work for the ESV development by precise comprehension of the vehicle dynamics factors, from the early stage of the development.

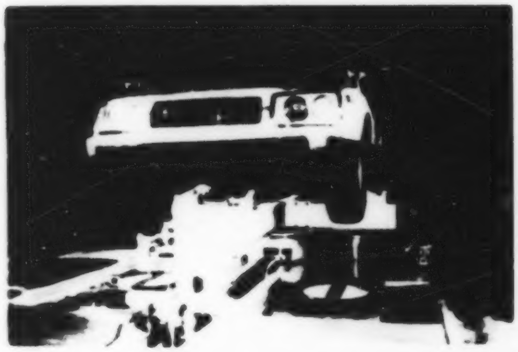

Figure 2

Tire characteristics, as well known, is an important factor affecting braking, handling and steering. We therefore manufactured several types of prototype tires and tested them.

Figure 3 shows how we measure tire characteristics.

Figure 4 shows the cornering force and self-aligning torque of a prototype tire. In the first series of the ESV tests, this tire was used as the standard.

Figure 5 shows the brake system used for the first series of tests of Toyota ESV. It consists of four principal subsystems, namely,

1. skid control system,
2. split system,
3. hydraulic booster,
4. 4-wheel disc brakes.

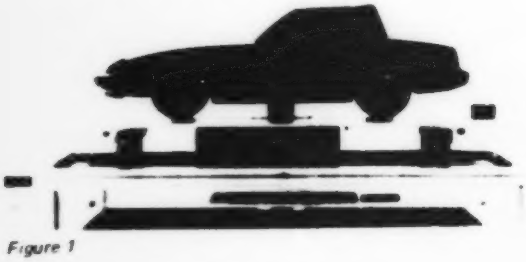

Figure 1

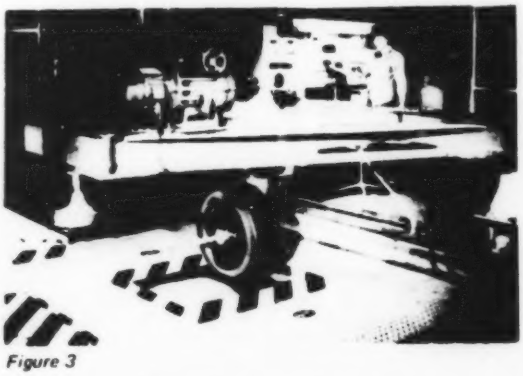

Figure 3

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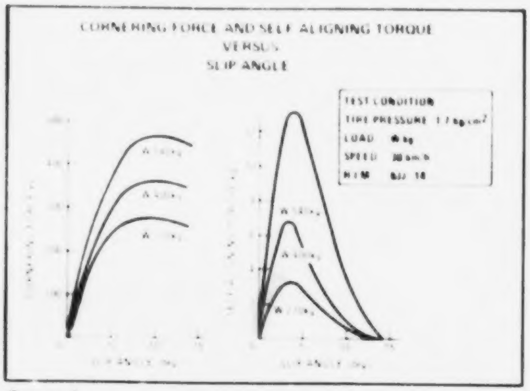


Figure 4

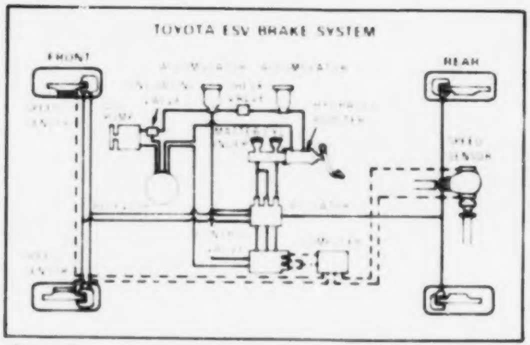

TOYOTA ESV BRAKE SYSTEM
FRONT
REAR
PUMP
HYDRAULIC BOOSTER
SPEED SENSOR
ACTUATOR

Figure 5

Figure 6 shows the skid control system where the four wheels are controlled by the signal of four-wheel speed-sensors.

Figure 7 compares the deceleration curves on dry, paved surface with and without the skid control system in operation. The difference in the stopping distance shows that the skid control system enables the vehicle to apply the full advantage of the surface friction.

Figure 8 shows the split system. The primary circuit applies to the 40% to the front wheel braking force, and the entire braking force to the rear wheels. The secondary circuit applies to the 60% to the front wheel braking force.

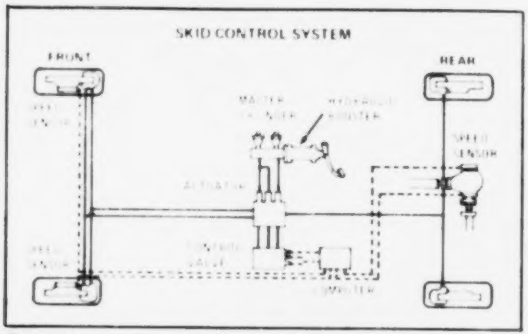

SKID CONTROL SYSTEM
FRONT
REAR
MASTER CYLINDER
HYDRAULIC BOOSTER
SPEED SENSOR
ACTUATOR
CONTROL VALVE
COMPUTER

Figure 6

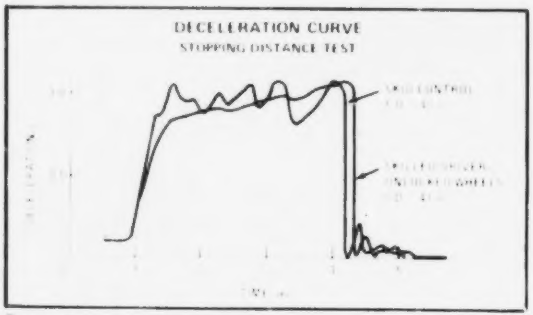


Figure 7

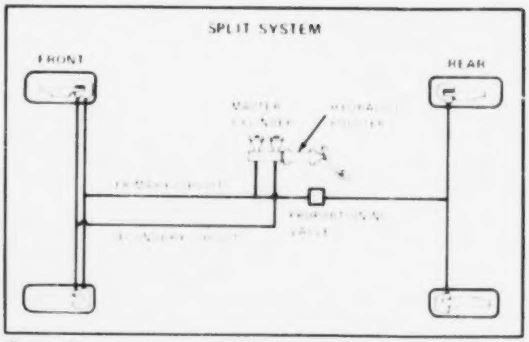

SPLIT SYSTEM
FRONT
REAR
MASTER CYLINDER
HYDRAULIC BOOSTER
PRIMARY CIRCUIT
SECONDARY CIRCUIT
PROPORTIONING VALVE

Figure 8

Figure 9 shows one of the several brake dynamometers in the Higashifuji Technical Center. This dynamometer is designed to measure the entire braking performance except the tire characteristics at the contact patch.

It thus enables precise and detailed observation regarding, for instance,

1. the braking efficiency for the optimum distribution in front and rear.

Figure 9

2. the balance of braking forces among the wheels.
3. the difference of braking force depending on various conditions.

Figure 10 shows the front-rear braking force distribution under normal conditions measured by this brake dynamometer. As indicated, the brake-force distribution is set very close to the ideal condition in case of simultaneous locking of the front and the rear wheels.

It can also be determined that the variation derived from temperature and braking history is very small. Within the region of braking force distribution shown in Figure 10, braking efficiency of 80% is assured theoretically.

Figure 11 shows the braking characteristics under partial failure conditions.

Because of the equipment of the split system, sufficient deceleration and stable braking characteristics are observed.

Toyota ESV is equipped with a hydraulic brake booster as shown in Figure 12, which enables it to provide compact and high-response power assist. The brake is sufficiently effective even in the event of booster failure.

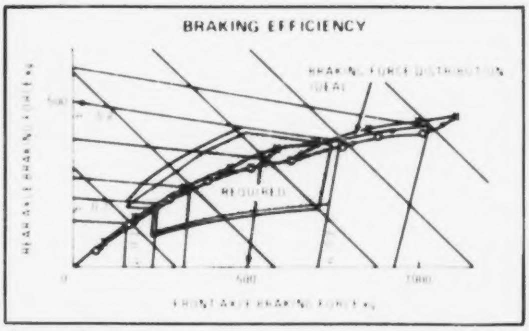


Figure 10

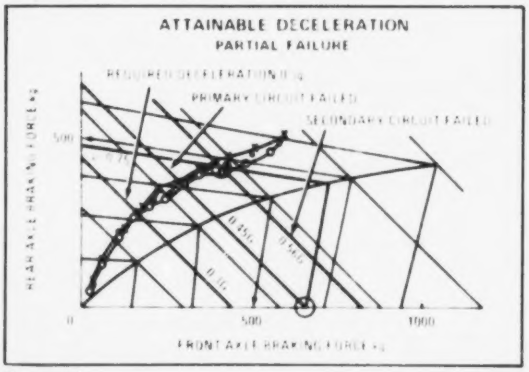


Figure 11

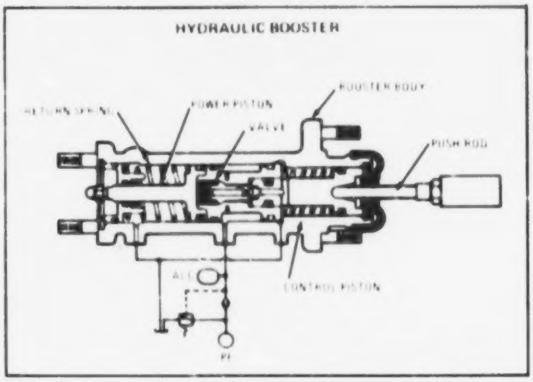


Figure 12

Figure 13 shows the pedal force vs. the deceleration obtained in vehicle tests with the booster normally operating.

Figure 14 shows the results of pedal force vs. deceleration curve with the booster failure.

The same with partial failure in Figure 15.

Figure 16 is a table, shown in a summarized form, showing the compliance to the Japanese ESV specifications, including other items discussed so far.

The first series of tests have revealed that Toyota ESV brake characteristics meet the Japanese ESV specifications. However, the items regarding the relations with other characteristics and the affections of occupant feeling are left as future studies.

The second series of tests will be concerned with these items: parking brake, improvement of each component and confirmation of their reliability.

I would now like to enter the discussion of handling and steering.

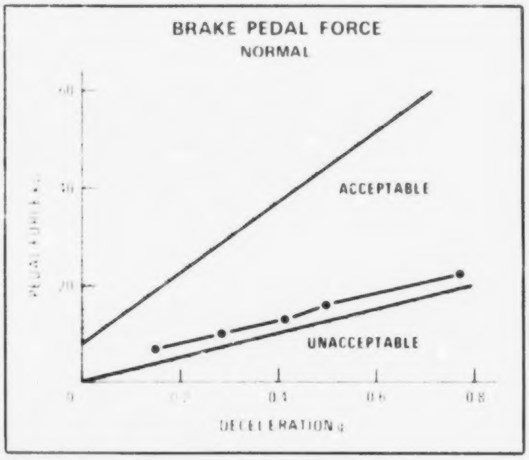


Figure 13

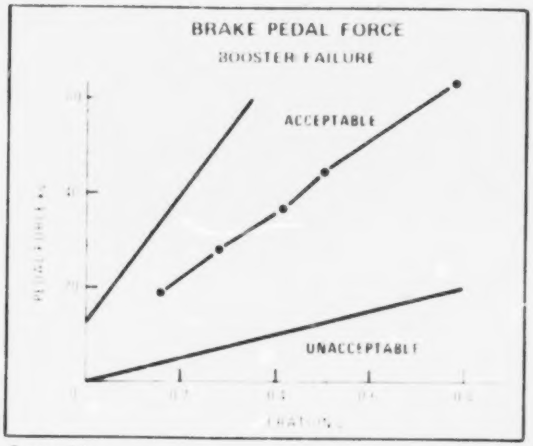


Figure 14

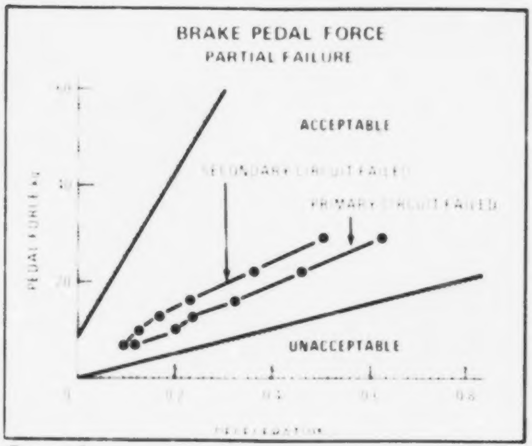


Figure 15

Figure 17 shows the steady-state yaw response characteristics. Case "A" is the test result of the original design for the first prototypes and case "B" is that of the design modification for research and development.

In case "A," the vehicle shows steering characteristics slightly closer to the neutral steer than the Japanese ESV specifications. We further carried out the tests with modified steering characteristics by changing tires and suspension properties.

BRAKE DATA SUMMARY		
PARAMETER	REQUIREMENT	PERFORMANCE
STOPPING DISTANCE (NORMAL)	50m MAX	41m
STOPPING DISTANCE (PARTIAL FAILURE)	120m MAX	64m
STOPPING DISTANCE (SERVO FAILURE)	105m MAX	51m
FADE	60kg MAX	13.2kg
RECOVERY	60kg MAX	8.5kg
PARKING BRAKE (UP HILL)	60kg MAX	———
PARKING BRAKE (DOWN HILL)	60kg MAX	———

Figure 16

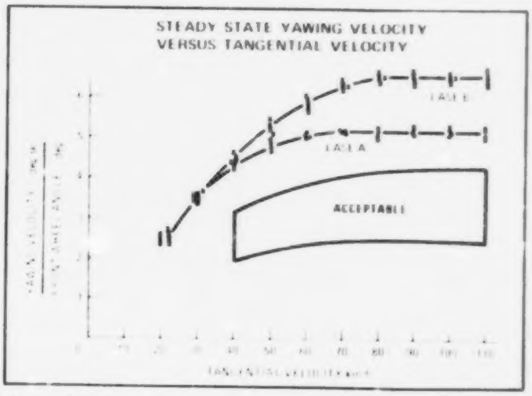


Figure 17

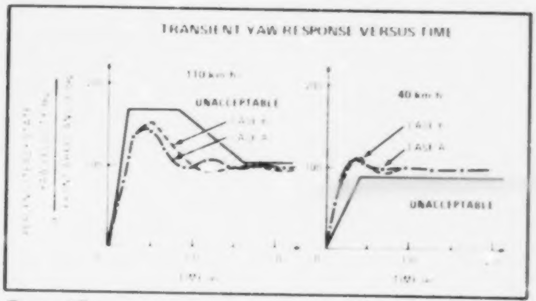


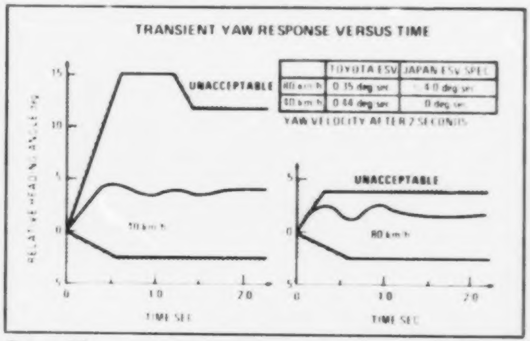
Figure 18

In our research, it was found that the characteristics like case "B" were rather agreeable to our driving feeling.

Figure 18 shows transient yaw response. The graph on the left side shows the case at 110 km/h and the one on the right side at 40 km/h.

Figure 19 shows returnability. The case of 40 km/h is shown on the left side and one of 80 km/h on the right. The table in the upper right corner shows the yaw velocity, two seconds after leaving the steering wheel.

Figure 20 shows the steering returnability process after releasing the steering wheel at 20 km/h. The



	TOYOTA ESV	JAPAN ESV SPEC
40 km/h	0.35 deg/sec	4.0 deg/sec
80 km/h	0.44 deg/sec	0 deg/sec

YAW VELOCITY AFTER 2 SECONDS

Figure 19

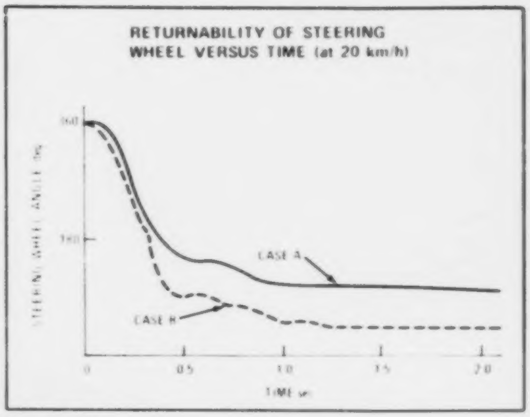


Figure 20

vertical coordinate shows the steering angle and the horizontal coordinate shows the time. Case "A" is the result with the original design while in case "B" frictions were reduced.

It has been found that the returnability can be improved. However, it is almost impossible physically to completely return the steering wheel to the neutral position, and enforcement of the above returnability would cause some adverse effects to other characteristics.

Figure 21 shows a circular turn used for measuring maximum lateral acceleration.

Figure 22 shows an example of instrumentation used for measuring handling and steering characteristics.

Figure 23 shows the test results for maximum lateral acceleration under various tire inflation pressures. In all of these cases, stable performances were obtained well over the value of 0.66 or 0.65 g of ESV specifications.

Figure 24 shows the control characteristics at breakaway. The Table at the bottom shows the sensitivity of the steering control. Both meet the Japanese ESV specifications sufficiently.

Figure 25 shows a test scene of crosswind sensitivity. The results are summarized in Figure 26.

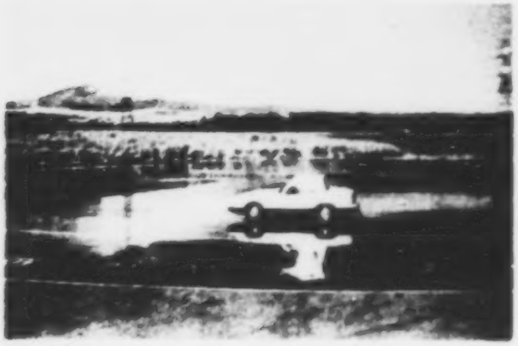


Figure 21

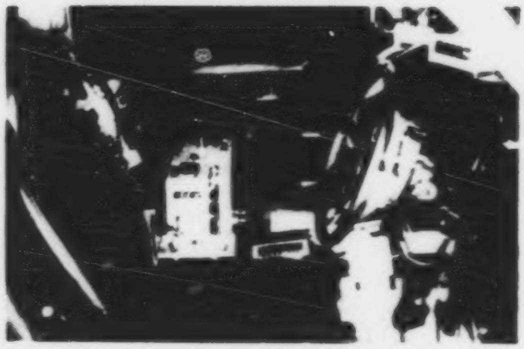


Figure 22

MAXIMUM LATERAL ACCELERATIONS

TIRE PRESSURES (kg/cm²)	LATERAL ACCELERATIONS TOYOTA ESV (G) CLOCKWISE	 COUNTER CW	 JAPAN ESV SPEC (G)
DESIGN VALUE (1.70)	0.81	0.81	0.65
120% DESIGN VALUE (2.04)	0.78	0.77	0.66
80% DESIGN VALUE (1.36)	0.77	0.80	
120% DESIGN FRONT (2.04) 80% DESIGN REAR (1.36)	0.82	0.82	
80% DESIGN FRONT (1.36) 120% DESIGN REAR (2.04)	0.78	0.77	

Figure 23

BREAKAWAY TEST RESULTS

RUN NO	JAPAN ESV SPEC LATERAL ACC (G)	STEERING WHEEL MOVEMENTS NECESSARY TO REGAIN 500 deg	RECOVERY TIME 4.0 sec
1	[illegible]	[illegible]	[illegible]
2	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
4	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]

STEERING CONTROL SENSITIVITY

VELOCITY (km/h)	TOYOTA ESV kg · cm/2 deg/sec	JAPAN ESV SPEC kg · cm/2 deg/sec
[illegible]	[illegible]	
[illegible]	[illegible]	

Figure 24

Figure 25

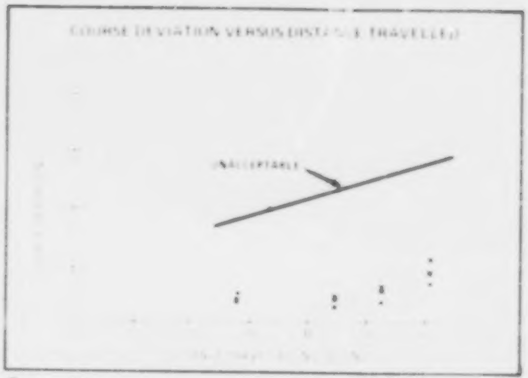
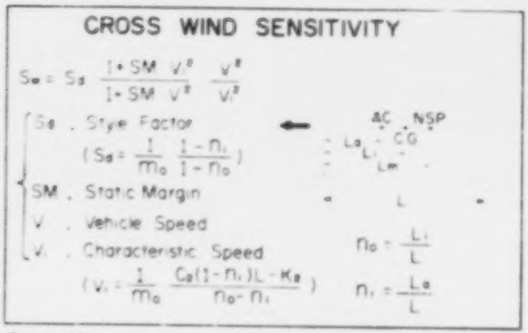


Figure 26



CROSS WIND SENSITIVITY

$S_w = S_d \frac{1+SM\ V_c^2}{1+SM\ V^2}\ \frac{V^2}{V_c^2}$

S_d . Style Factor

$\left(S_d = \frac{1}{m_0}\ \frac{1-n_1}{1-n_0} \right)$

SM . Static Margin

V . Vehicle Speed

V_c . Characteristic Speed

$\left(V_c = \frac{1}{m_0}\ \frac{C_0(1-n_1)L-K_0}{n_0-n_1} \right)$

$n_0 = \frac{L_1}{L}$

$n_1 = \frac{L_a}{L}$

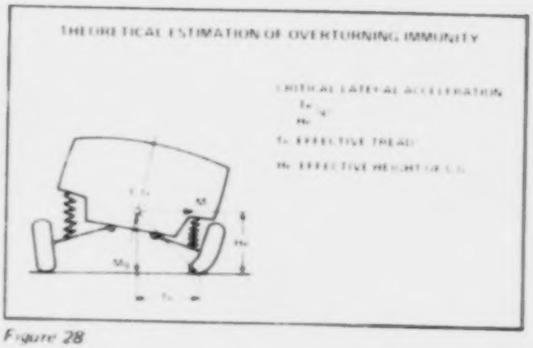
AC
NSP
CG
La
Li
Lm
L

Figure 27

The influence of the crosswind under fixed control can be expressed by the equation in Figure 27, where Sw means coefficient of crosswind sensitivity.

Sd is a constant value determined by the said aeronautical characteristics and the basic specifications of the vehicle. Using this method we could obtain good results in the viewpoint of making the vehicle crosswind stability satisfactory from the planning stage.

Figure 28 shows our approach to the rollover immunity.



THEORETICAL ESTIMATION OF OVERTURNING IMMUNITY
CRITICAL LATERAL ACCELERATION
Te EFFECTIVE TREAD
He EFFECTIVE HEIGHT OF C.G.
He
Te

Figure 28

Figure 29

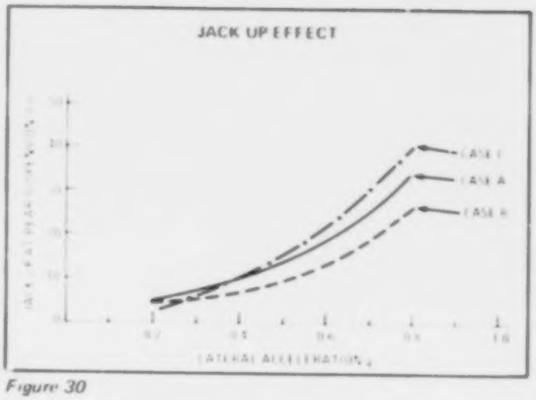


Figure 30

The critical lateral acceleration *a* calculated based on the original design is as follows.

$$a = 1.13\,g$$

Figure 29 shows a J-turn test at 110 km/h. In our test, no rollover occurred in any case.

Many factors are involved in the effective tread Te and the effective height He of the center of gravity that decide the critical lateral acceleration *a*. In these factors, jacking-up effect is a very important one.

Figure 30 shows the amount of jack-up for three kinds of suspension properties. Case "A" is the result with the original design. Case "B" and "C" are the results with modified suspension characteristics. Such modifications influence the axle steer characteristics and eventually handling and steering.

As described so far, the original design characteristics used for the series of tests could meet almost all the specifications of the Japanese ESV. It is easy to make a drastic improvement to each of these characteristics individually, but it is not easy to improve the overall performance of the vehicle.

We intend to carry out the tests, including the items left undone and research from the standpoint of whole vehicle systems, in the next stage of ESV development.

We also consider that the study of man-vehicle system is as important as these characteristics. During these studies, it seems that some appropriate modification of the ESV specifications may become necessary. Nevertheless the tests conducted so far will prove to be a great assistance for producing safer vehicles in the future, and we will continue efforts for such a development.

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART 1

ENERGY MANAGEMENT SYSTEM

AKIHIRO WADA, *Manager*
Body Development Section
Body Designing Department
Toyota Motor Company, Ltd.

After the tests on the modified cars and the first prototypes, we undertook a second series of prototypes. I will try to describe to you the body structure, test results and the feasibility for meeting each of the specifications.

The Front Body Structure

Figure 1 shows the relationship between the crush distance and the compartment deceleration levels at the 80km/h frontal barrier impact.

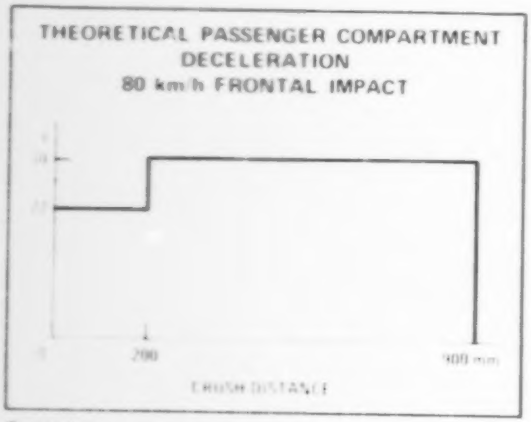


Figure 1

It is designed so that 200 mm of total crush distance should be consumed by the bumper shock absorbers and the remaining 700 mm should be consumed by the body deformation. The characteristics of the bumper shock absorber is so set that the deceleration of the vehicle is 22 g. at the 80 km/h barrier impact. This is the result of an attempt to keep the weight of the bumper absorbers and mounting portion within a reasonable limit.

As shown in Figure 2, the silicone rubber-type bumper shock absorbers generate 6 g at the 15 km/h barrier impact. Silicone rubber type was chosen because of the small size and weight, as well as easy control of the load versus the impact speed, although at first a hydraulic-pneumatic type also was tried.

The front bumper, in order to cope with the 80 km/h pole impact, is designed to withstand a load corresponding to about 25 g, slightly higher than the load in the shock absorbers, by utilizing 80 kg/mm^2 high tensile strength steel, resulting in the weight of 27 kg. For the surface of the bumper, foamed polyurethane cover is used.

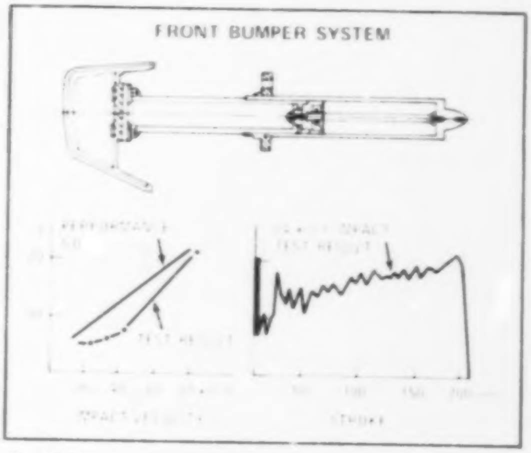


Figure 2

Figure 3 shows the front body structure. Except for the rear portion, the chassis frame takes the form of the perimeter frame for our mass-production car "Crown."

The side rails are fastened to the lower surface of the locker panels by bolts without rubber insulators, in order to distribute to the various portions of the vehicle the large impact force on the frame. The energy-absorbing members are provided in the upper part of the front fender aprons in order to prevent pitching during impact. These members have an undesirable side-effect of deforming the door and the door opening structure at impact, and these are designed to generate only the minimum force needed. The ratio of force generated by these members and that by the chassis frame is 2:8.

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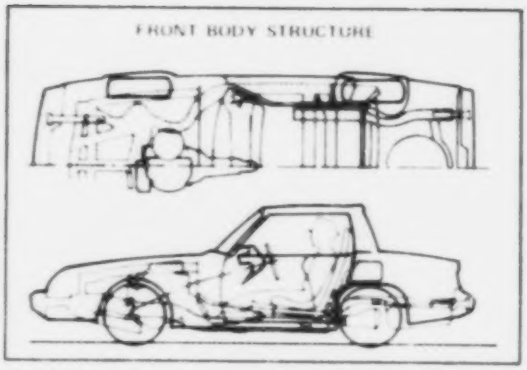


Figure 3

As shown in Figure 4, there are two types of frame which serve as the main energy absorber: one with "S" shaped energy-absorbing members and the other based on the mass-production model "Crown," but with additional members for greater energy-absorption characteristics. These two types were designed and trial-manufactured almost simultaneously.

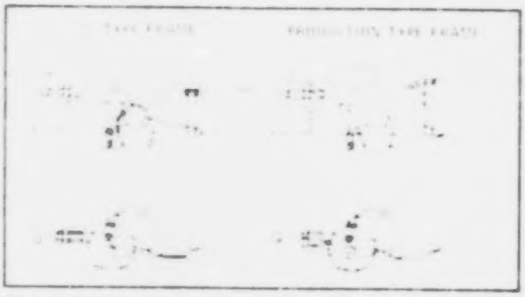


Figure 4

Figures 5 and 6 show the results of the 80 km/h flat barrier impacts of modified vehicles with "S" type and mass-production type frames. The "S" type frame was installed to the modified car, while the mass-production type frame was installed to a dolly

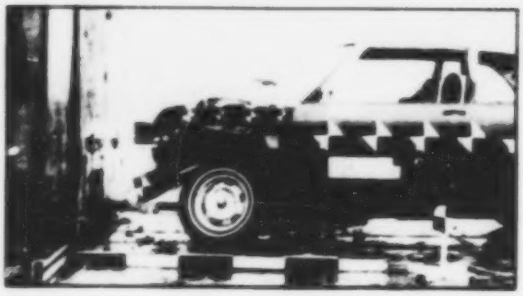


Figure 5

with equivalent properties. We found good results in both cases, judging from the loads on the barrier.

In terms of weight, the production-type frame appeared to be capable of absorbing a little more energy, but the difference was so slight that the "S" type too can easily attain the same result by modification. Figures 7 and 8 show the results of the 80 km/h pole barrier impacts of modified cars with the two types of frames.

Since in this case, too, only a slight difference was observed, we decided to use only the mass-production

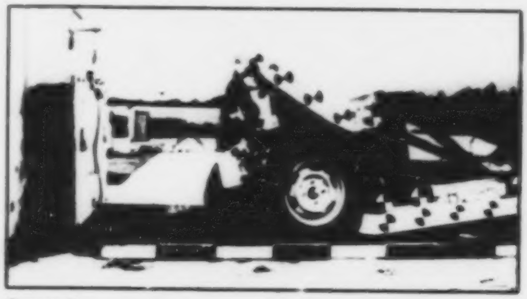


Figure 6

Figure 7

Figure 8

frame, which is easier to produce in light of future convenience. This, of course, does not eliminate the need for using high-tensile strength steel of 60-80 kg/mm^2 for the main portion of the frame, to reduce weight in optimum balance between the various parts. This balance must consider the characteristics of the high-tensile strength steel, such as tensile strength, elongation, formability, and weldability. High-tensile strength steel is used also for the front pillars, door beams, etc., leaving such problems as cost of material, difficulty of stamping and welding.

As shown in Figure 9, the hood is designed to bend above the front wheels at an impact by providing a weak portion in the hood.

In order to prevent the hood end from intruding into the windshield, hooks are provided at both sides of the hood, so that its upward and rearward movement is restrained. The reliability of the hood hooks is not sufficiently high in the first prototype, but will be improved in the second prototype. Moreover, with adoption of concealed wipers, the bottom edge of the windshield is located high to prevent the hood intrusion.

Figure 10 shows the result of the 80 km/h flat-barrier impact test of the first prototype.

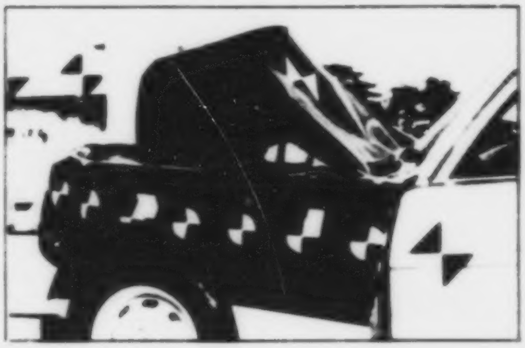


Figure 9

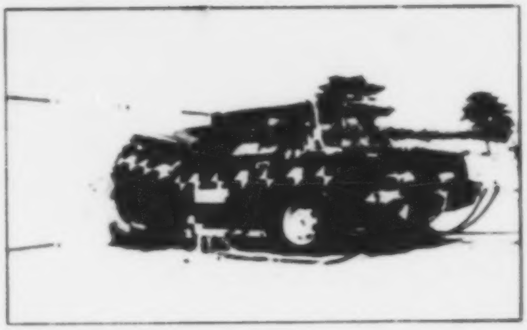


Figure 10

As shown in Figure 11, the average floor deceleration was 27 g, the total crush distance 710 mm, and the intrusion into the passenger compartment 150 mm.

Figure 12 shows results of the 80 km/h pole barrier impact test of the first prototype.

As shown in Figure 13, the average floor deceleration was 24 g, the total crush distance 1,000 mm, and the intrusion into the passenger compartment 160 mm. In both tests, the doors on both sides could easily be opened.

The Rear Body Structure

The rear body structure, similar to ordinary frameless bodies, features the "S" type side members having a large cross section area as illustrated in Figure 14, where absorption of most of the energy is made by means of the deformation of these side members.

The fuel tank and spare wheel are located at different heights to avoid colliding with each other.

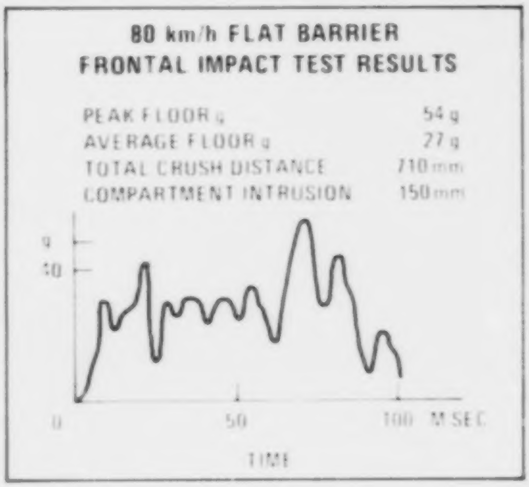


Figure 11

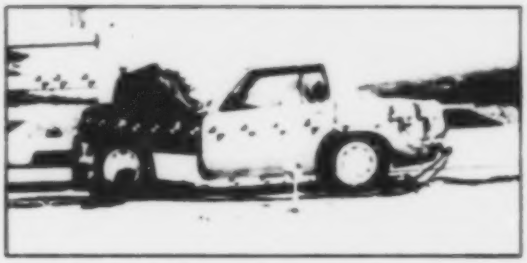


Figure 12

The silicone-rubber type bumper shock absorbers have a stroke of 100 mm, generating 6 g at the 15 km/h and 30 g at the 80 km/h barrier impact. The total crush distance including the shock absorber stroke is designed to be 600 mm, at the 80 km/h rear impact by a 4,000 pound moving barrier. Figure 15 shows the result of such an impact test with a first prototype. The test data are tabulated in Figure 16.

The peak floor deceleration was 94.2 g, the average floor deceleration 22 g, and the total crush distance 590 mm.

But the occupant crash protection requirements of the 80 km/h rear impact has not been entirely met. For example, the passenger's head shows 81.7 g for no more than 3m sec., and the chest 66 g. This has been caused by the passenger being pushed forward by an intrusion behind the seats, which was a little greater than scheduled, and will be avoided by controlling the strength of the side members.

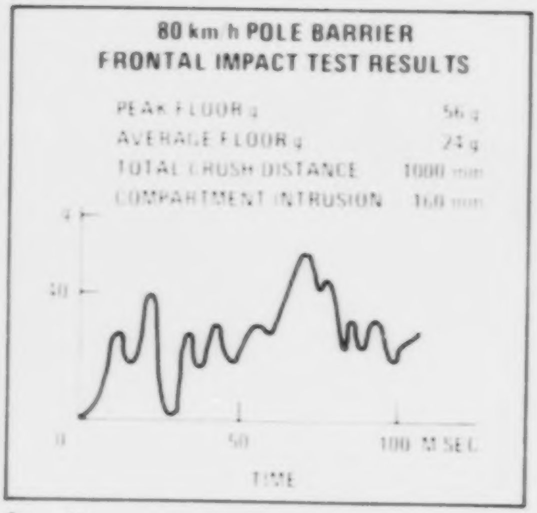


Figure 13

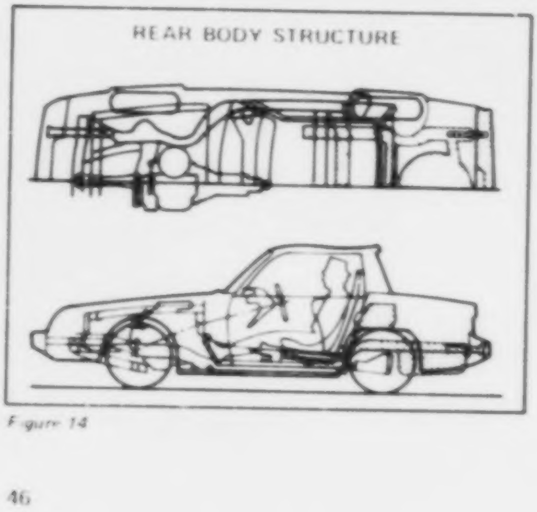


Figure 14

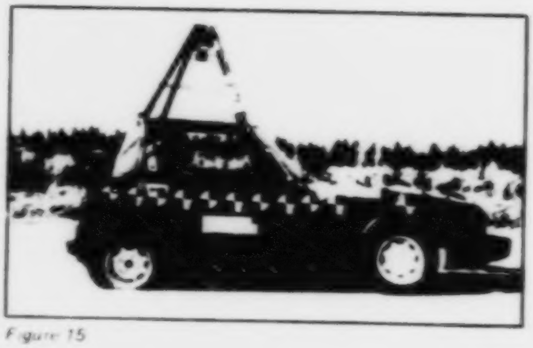
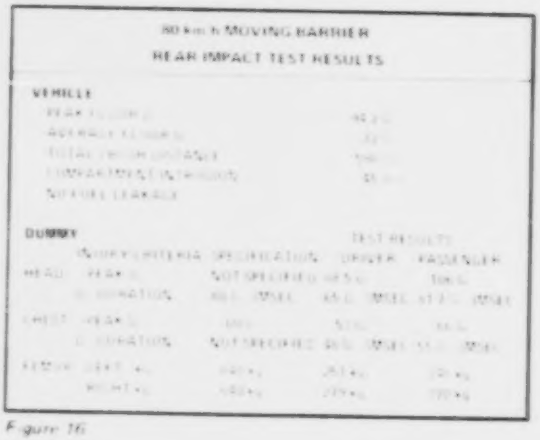


Figure 15



80 km/h MOVING BARRIER
REAR IMPACT TEST RESULTS

VEHICLE

PEAK FLOOR g	94.2 g
AVERAGE FLOOR g	22 g
TOTAL CRUSH DISTANCE	590 mm
COMPARTMENT INTRUSION	[illegible]
NO FUEL LEAKAGE	

DUMMY

	INJURY CRITERIA	SPECIFICATION	TEST RESULTS DRIVER	TEST RESULTS PASSENGER
HEAD	PEAK g	NOT SPECIFIED	[illegible]	[illegible]
	G DURATION	80 G 3MSEC	[illegible] 3MSEC	81.7 G 3MSEC
CHEST	PEAK g	60G	[illegible]	66 G
	G DURATION	NOT SPECIFIED	[illegible] 3MSEC	[illegible] 3MSEC
FEMUR	LEFT kg	[illegible] kg	[illegible] kg	[illegible] kg
	RIGHT kg	[illegible] kg	[illegible] kg	[illegible] kg

Figure 16

The Side Body Structure

Figure 17 shows the side body structure. Inside the door there is installed a door beam of 100 kg/mm² high-tensile strength steel, to prevent intrusion.

Further occupant protection is provided by an energy-absorbing rigid polyurethane foam in the door trim and a seat divider with a steel metal core and a vinyl-skinned non-rigid polyurethane foam on the surface. In the case of the 50 km/h car-to-car side impact, the force generated by the shock absorbers was set lower than the door beam strength, in order to reduce the door weight.

Figure 18 shows the 50 km/h car-to-car side impact test against the driver side.

As shown in Figure 19, the peak floor deceleration of the struck car was 24 g, and the compartment intrusion 57 mm. The deceleration at the driver's head was 30.6 g, the chest 36.9 g, the passenger's head 69.9 g and the chest 29.7 g. All values meet the specifications.

Figure 20 shows the 25 km/h pole side-impact test against the passenger side.

As shown in Figure 21, the peak floor deceleration was 15 g, and the compartment intrusion 125 mm. The deceleration at the driver's head was 19.7 g, the chest 29.1 g, the passenger's head 23.6 g and the chest 24.8 g.

All values meet the specifications.

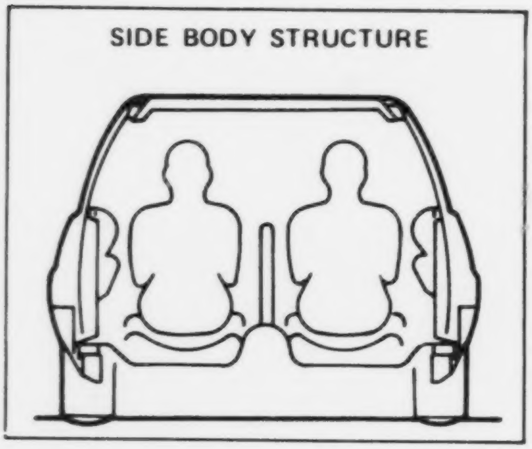


Figure 17

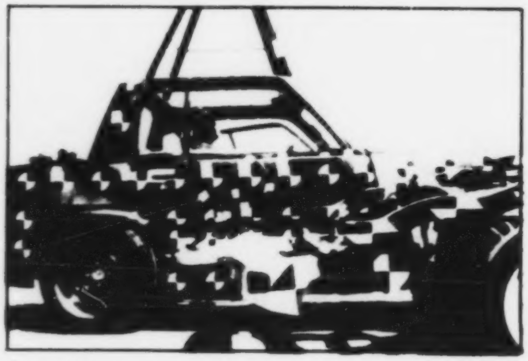


Figure 18

50 km/h CAR TO CAR
SIDE IMPACT TEST RESULTS

VEHICLE				
PEAK FLOOR G OF IMPACTED CAR			25 G	
COMPARTMENT INTRUSION			57 mm	
			TEST RESULTS	
DUMMY			DRIVER	PASSENGER
	INJURY CRITERIA	SPECIFICATION	(IMPACTED SIDE)	
HEAD	PEAK G	NOT SPECIFIED	30.6 G	69.9 G
	G DURATION	80 G 3MSEC	MET	MET
CHEST	PEAK G	60 G	36.9 G	29.7 G
	G DURATION	NOT SPECIFIED		

Figure 19

The Roof Structure

Figure 22 shows the 50 km/h inclined dolly test using one of the first prototypes.

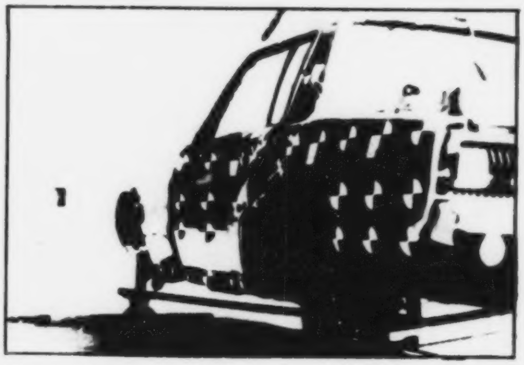


Figure 20

25 km/h POLE BARRIER
SIDE IMPACT TEST RESULTS

VEHICLE				
PEAK FLOOR G			15 G	
COMPARTMENT INTRUSION (DOOR CENTER)			125 mm	
			TEST RESULTS	
DUMMY			DRIVER	PASSENGER (IMPACTED SIDE)
	INJURY CRITERIA	SPECIFICATION		
HEAD	PEAK G	NOT SPECIFIED	19.7 G	23.6 G
	G DURATION	80 G 3MSEC	MET	MET
CHEST	PEAK G	60 G	29.1 G	24.8 G
	G DURATION	NOT SPECIFIED		

Figure 21

Figure 22

Figure 23 tabulates the test data for this vehicle. The compartment intrusion was 126 mm at the front pillar portion. The deceleration at the driver's head was 46.2 g for no more than 3 m sec., the chest 17.9 g, the passenger's head 106 g for no more than 3 m sec., and the chest 24.4 g. The high g of passenger head was caused by the head striking the roof with insufficient padding. In the second prototype, this point will be improved. But we would add that the results of a rollover test must be read with some uncertainty in terms of repeatability.

Occupant Protection Systems

The occupant protection at frontal impact is provided by the passive lap belt and the gas bag restraint systems. Figure 24 shows the passive lap belt systems.

The passive lap belt has an electrical emergency-locking retractor located at the rearmost anchorage portion of the door. It fastens when the door closes and releases on its opening by means of a hook that moves along a rail attached to the door's inside trim.

Since the retractor locks at 0.5 g vehicle deceleration, the belt secures the occupants in position at panic braking and sharp turns. In the event of a crash, it protects the occupants in coordination with the instrument panel knee pad.

For the gas bag sensor, the pre-crash type Toyota Radar Sensor was used, for the reason of sufficient inflation time and large ride-down effects. Figure 25 shows the time budget for the 80 km/h barrier impact.

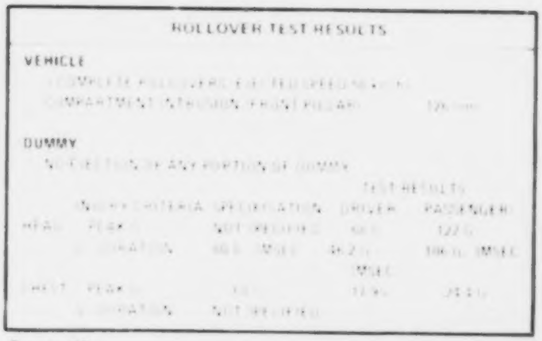
The radar sensor signals the actuator 65 m sec. before the impact, and the bags completely deploy 5 m sec. after impact. Calculations based on the test results disclosed the ridedown effects of 15-25%, as expected.

For low-speed impact under 30 km/h, the gas bags are not deployed. The passenger is restrained by the large instrument panel of the padded crushable sheet metal structures. Figure 26 shows these features.

Figure 27 shows the result of a sled test simulating the 30 km/h frontal barrier impact, barely below the compliance level.

The margin will be easily improved by making the instrument panel more susceptible to deformation.

The greatest trouble in the 80 km/h impact is the intrusion of the engine into the passenger compartment, resulting in the deformation of the steering column support system. This intrusion lifted the steering column shaft and, therefore, the energy-



ROLLOVER TEST RESULTS

VEHICLE

[illegible]

COMPARTMENT INTRUSION (FRONT PILLAR) 126 mm

DUMMY

NO EJECTION OF ANY PORTION OF DUMMY

	INJURY CRITERIA	SPECIFICATION	TEST RESULTS DRIVER	TEST RESULTS PASSENGER
HEAD	PEAK G	NOT REQUIRED	[illegible]	122 G
	G DURATION	80 G 3MSEC	46.2 G 3MSEC	106 G 3MSEC
CHEST	PEAK G	[illegible]	17.9 G	24.4 G
	G DURATION	NOT REQUIRED		

Figure 23

Figure 24

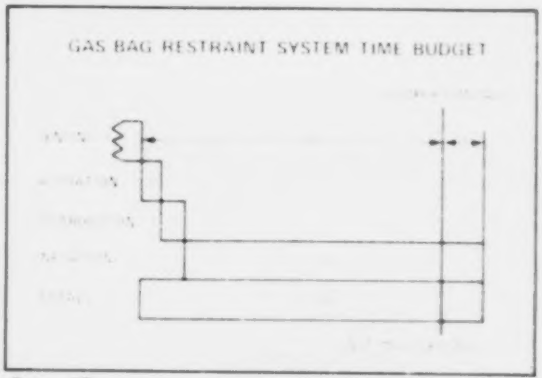


Figure 25

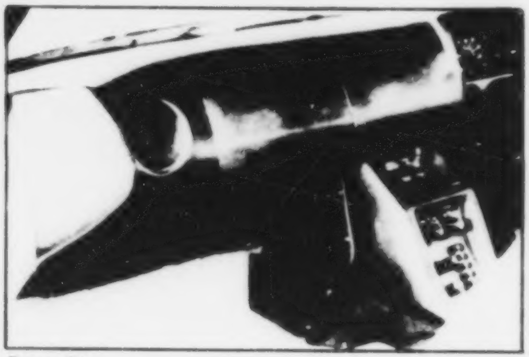


Figure 26

GAS BAG NON DEPLOYMENT SLED TEST RESULTS

(30 km/h EQUIVALENT BARRIER IMPACT)

INJURY CRITERIA		SPECIFICATION	DRIVER	PASSENGER
HEAD	PEAK G G DURATION	NOT SPECIFIED 80 G 3MSEC	[illegible] [illegible] 3MSEC	76 G 75G 3MSEC
CHEST	PEAK G G DURATION	60 G NOT SPECIFIED	41 G 40 3MSEC	59 G 56G 3MSEC
FEMUR	RIGHT kg LEFT kg	640 kg 640 kg	215 kg 285 kg	367 kg 105 kg

Figure 27

absorbing stroke did not function effectively during dummy contact. So we changed the support system as shown in Figure 28.

This could be improved by limiting the effect of the engine intrusion. As shown in Figure 29, the occupant crash-protection requirement of the 80 km/h impact has not been entirely satisfied.

From these test results the passenger's head, chest and femur values meet the specifications to the flat barrier and the pole impact. The driver's head shows 60 g for no more than 3 m sec., and the chest 65 g to the flat barrier impact. In the pole impact the head 52 g for no more than 3 m sec., the chest 60 g. Some values exceed the specifications. There still remain various unresolved problems, such as deformation characteristics of the energy-absorbing steering column, the gas bag volume and the internal pressure control.

Our ESV has been gradually improved in many respects. However, we have not yet met the specifications regarding some of the injury criteria and the vehicle weight. The dummy's injury criteria would be attainable by improving the restraint system, but the weight reduction would be difficult, considering the large weight excess, in spite of the abundant use of high-tensile strength steel in the frame, pillar, door beam, bumper, etc. Although we try to meet all the requirements, we find it difficult to observe this weight limit.

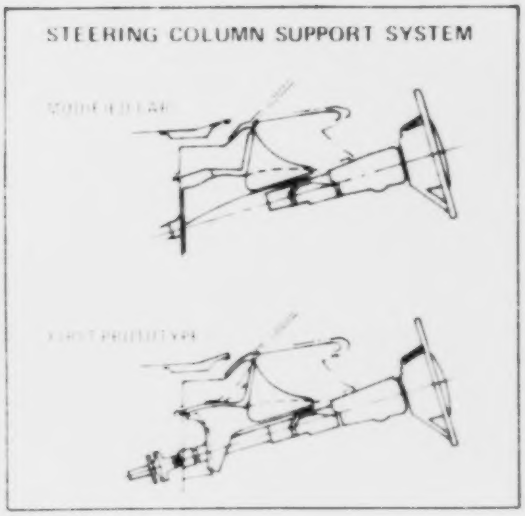


Figure 28

[illegible] FRONTAL IMPACT TEST RESULTS

		FLAT BARRIER IMPACT		POLE IMPACT	
INJURY CRITERIA	SPECIFICATION	DRIVER	PASSENGER	DRIVER	PASSENGER
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Figure 29

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO PART 1

SUMMARY OF TOYOTA ESV DEVELOPMENT

JIRO KAWANO, *Manager*
Technical Chief Designer, ESV
Toyota Motor Company, Inc.

In Washington, I reported mainly on the composition of our ESV. Today I will describe the results of development and evaluation on our first prototypes.

As far as braking performance is concerned, Toyota ESV is equipped with a newly developed, high-performance service brake system composed of dual-circuit, four-wheel discs, anti-skid system and a hydraulic booster. It meets all the requirements, as shown in Figure 1.

For the parking brake system, various systems are under development and we anticipate no problem whatever that the system will be adopted.

For steering characteristics, Toyota ESV is optimized for its short wheelbase, unlike the requirements of the Japanese ESV specifications, so that it has near neutral characteristics regarding steady-state yaw response, as the engineer in charge explained previously.

The handling performance of our ESV meets all the requirements of Japanese ESV specifications as Figure 2 shows.

We particularly aimed for, and attained, the overturning immunity at 110 km/h, specified as a target in Japanese ESV specifications.

Most of the visibility requirements were incorporated at the vehicle planning stage. The details are shown in Figure 3.

It has been confirmed that most of the requirements have been met, although a few tests have not been completed. A couple of devices rated "preferable to equip" in the Japanese ESV specifications are not installed in the first prototypes.

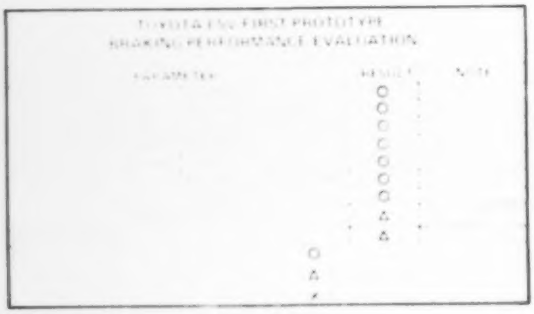
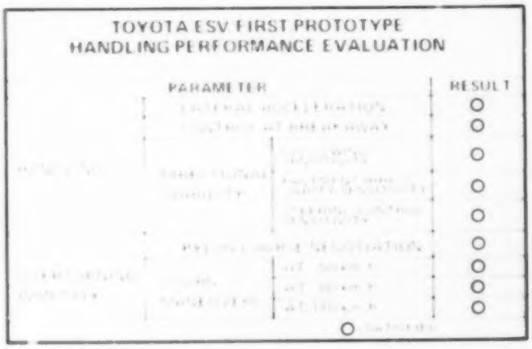

TOYOTA ESV FIRST PROTOTYPE
BRAKING PERFORMANCE EVALUATION

Figure 1


TOYOTA ESV FIRST PROTOTYPE
HANDLING PERFORMANCE EVALUATION

PARAMETER | RESULT

Figure 2

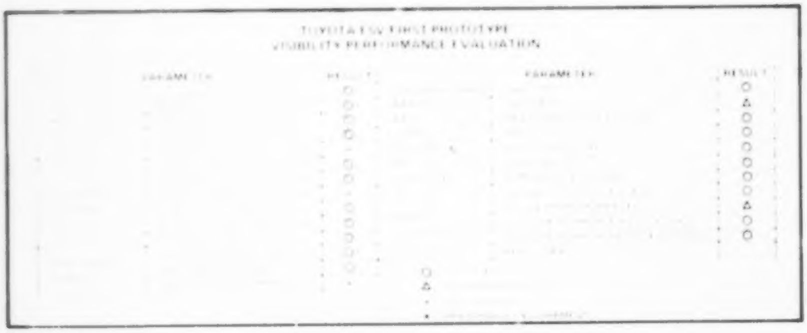

TOYOTA ESV FIRST PROTOTYPE
VISIBILITY PERFORMANCE EVALUATION

Figure 3

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I would like to move on now to the most critical requirements — crash injury reduction. We planned and developed our ESV on the premise that it should meet all the Japanese ESV specifications, including the target. I will explain the state-of-the-art.

Figure 4 is a compliance summary of vehicle structural crashworthiness. It has been confirmed that the requirement regarding the strength, rear end collision, side collision and rollover are almost met, despite some evaluations not yet completed, due to limited number of prototypes.

For the 80 km/h head on collision, the pole impact in particular, the desired results have not yet been entirely obtained as to the compartment integrity and/or intrusion, and further study will be made.

The occupant protection requirements, that is to say the deceleration and force at certain portions of the dummy or its ejection, are explained in Figure 5.

The unattained target is the protection of the driver at the 80 km/h head-on collision, and we are pouring efforts into its study.

Among the areas for further study are: the irregular behavior of the energy-absorbing steering column, due to unfavorable deformation of its supporting members, the unstable tendency of the inflated gas bag on the steering wheel, and the optimization of the energy-absorption properties. Except for these points, we believe we have satisfied the specifications.

We at Toyota, have designed and fabricated an ESV to meet the Japanese ESV specifications with success in many ways. Except for the weight problem, the points which are not yet satisfactory regarding the few requirements of crashworthiness will have to be completed by the time our ESVs are delivered to our government.

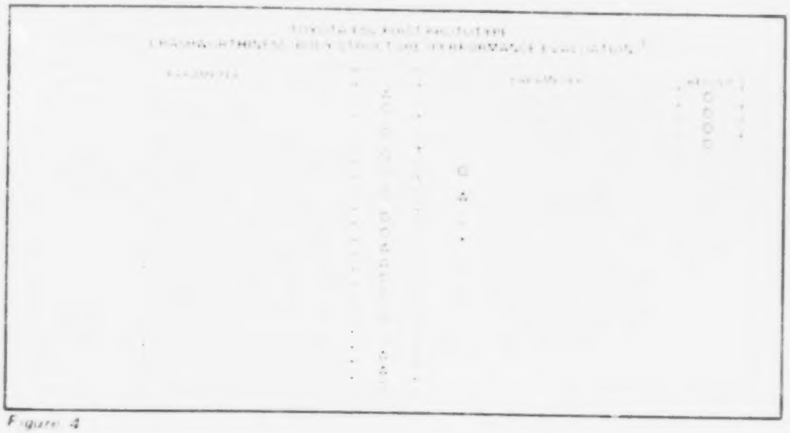
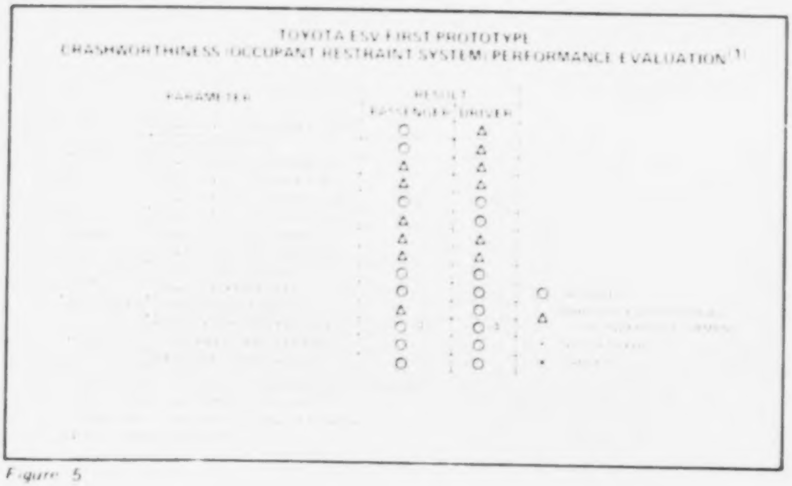


Figure 4



TOYOTA ESV FIRST PROTOTYPE
CRASHWORTHINESS (OCCUPANT RESTRAINT SYSTEM) PERFORMANCE EVALUATION[1]

PARAMETER	RESULT	
	PASSENGER	DRIVER
[illegible]	○	△
[illegible]	○	△
[illegible]	△	△
[illegible]	△	△
[illegible]	○	○
[illegible]	△	○
[illegible]	△	△
[illegible]	△	△
[illegible]	○	○
[illegible]	○	○
[illegible]	△	○
[illegible]	○	○
[illegible]	○	○
[illegible]	○	○

○ [illegible]
△ [illegible]
• [illegible]
• [illegible]

Figure 5

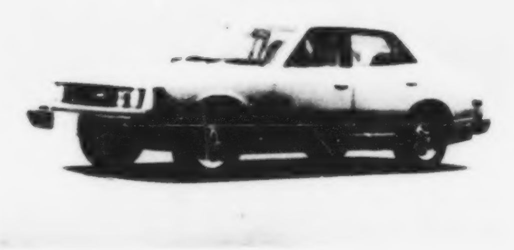
THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART 1

INTRODUCTION

SHOJIRO FUJITA, *Director,*
General Manager
Central Engineering Laboratory
Nissan Motor Company, Ltd



Since the beginning of 1971 when the Nissan ESV project team was established, we have, in cooperation with the companies of the Nissan Group, expended 2 billion yen and 300 man-years on this project (Figure 1). We have conducted basic tests on 74 production cars, 120 modified cars and 38 newly-modeled cars. It is expected that ten of the newly-manufactured Nissan ESVs will be delivered to the Government of Japan by the end of this year.

In January 1971, the project team was transferred from the Design Department to the Central Engineering Laboratories where, for the purpose of conducting comprehensive research and experiments, it has been assigned to the newly-established Safety Research Department.

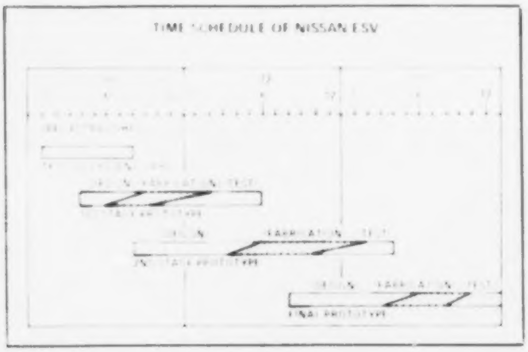


Figure 1

The Nissan ESV started with air bags and new monocoque body construction. It was then provided with two additional units — a seat belt and an energy-absorbing broach installed on the front end of the body. The former was called E_1, and the latter E_2. At the same time, we performed tests on modified model cars containing various possibilities we encountered in our research. Thus, we are obtaining a considerable amount of know-how, and are developing many related units. In System E_1, the vehicle is designed to be 1,250 kg in total weight and 4,415 mm in total length. In other words, it is 200 kg heavier and 200 mm longer than the Nissan Datsun 610.

With both Systems E_1 and E_2, some problems remain to be solved concerning the injury values of the driver's seat occupant in 80 km/h frontal barrier collisions. In tests carried out with actual vehicles, we have often obtained values complying with the injury criteria. However, when the reliability of the occupant protection system utilized is taken into account, we cannot say we have completely succeeded in meeting the standards.

The remaining specifications have all been satisfied. As to rear-end collisions, the vehicle speed has been rationally specified as 65 km/h, as we proposed during the previous International Conference. We consider the vehicle styling to be pretty well arranged. However, this ESV manufactured with specific quality control can display notable crashworthiness only under optimum conditions. We feel it would be impossible to obtain such performances during actual mass production. In spite of our considerable effort, the cost, practicability and productivity are not yet acceptable to us. In other words, we regard the Nissan ESV exclusively as a special solution to the ESV specifications, and at present we do not consider it realistic to require such a small-sized ESV to withstand 80 km/h collisions.

Using a modification of the Nissan ESV, we have concurrently developed and tested a lighter vehicle for 65 km/h frontal collisions. This model has obtained values complying with the ESV specifications, but it still has many shortcomings as a car for practical use. It is, however, considered to be a more appropriate objective for our research. In consideration of air pollution, traffic congestion and shortage of resources in the future, we should manufacture light-weight, small-sized vehicles which place less financial burden on the customer.

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART I

DEVELOPMENT OF THE NISSAN ESV

NAGAYUKI MARUMO, *Deputy Manager*
TERUO MAEDA, *Manager*
Safety Research Department
Nissan Motor Company, Ltd.

OUTLINE OF THE NISSAN ESV

Theme For Study Of Nissan ESV

The Nissan ESV, as a 4-door family sedan for four passengers, has been developed with stress placed on all-around safety in present traffic conditions. Therefore, instead of manufacturing a safe car with dimensions and weight increased, we have made efforts to produce an ESV with both the prescribed performance and dimensions limited to those of so-called small cars, which are most popular in Japan at this time (Figure 1).

Further efforts have been made to insure that the Nissan ESV has excellent handling performance, which is the advantage of small cars, and that it is provided with outstanding accident avoidance characteristics superior to the ESV specifications. As to crashworthiness, it is designed to secure safety in an 80 km/h frontal collision with a fixed barrier, a 65 km/h rear-end collision against a 1,800 kg (4,000 lbs) moving barrier, and a 50 km/h side collision with a car of the same weight.

We wish to make specific mention that we have made efforts related to pedestrian safety, in view of accidental pedestrian deaths accounting for as much as 35% of the traffic accident rate in Japan, due to road congestion.

Vehicle System Of The Nissan ESV

In order to keep the occupant safe in a frontal collision (Figure 2), it is necessary that in the initial stage of the crash, the occupant should be given deceleration in order to absorb his kinetic energy as much as possible by so called "ride down." The Nissan ESV has a crush stroke set at 850 mm in the frontal body during an 80 km/h frontal collision. With an objective that within 25 milliseconds after the crash the occupant begins to be restrained, and then undergoes deceleration within the injury criteria, we have developed the body construction and the occupant protection system, respectively.

MAIN SPECIFICATIONS OF NISSAN ESV	
TYPE	4 DOOR SEDAN
PASSENGERS	4
CURB WEIGHT	1,250 kg (TARGET 1,150 kg)
LENGTH	4,415 mm
WIDTH	1,640 mm
HEIGHT	1,385 mm (WITHOUT PERISCOPE)
WHEEL BASE	2,550 mm
TREAD	1,360 mm (FRONT & REAR)
ENGINE PISTON DISPLACEMENTS	1,770 cc

Figure 1

To limit the vehicle acceleration to approximately 20 g in 65 km/h rear-end collisions with a moving barrier (Figure 3), the crush stroke is set at 500 mm. With the standard compartment dimensions patterned after those of the Datsun 610 class, the Nissan ESV dimensions (Figure 4) are set to include a total length of 4,415 mm. If protection of the occupant in a side collision is to be done with padding alone, the padding should be extremely thick. To avoid this, we have allowed maximum vehicle body deformation within the limit of securing occupant survival space, in order to reduce the body deceleration to a minimum, to increase "ride down," and to make the padding thin. Utilizing this idea, we have succeeded in minimizing the vehicle body width to 1,640 mm (Figure 5).

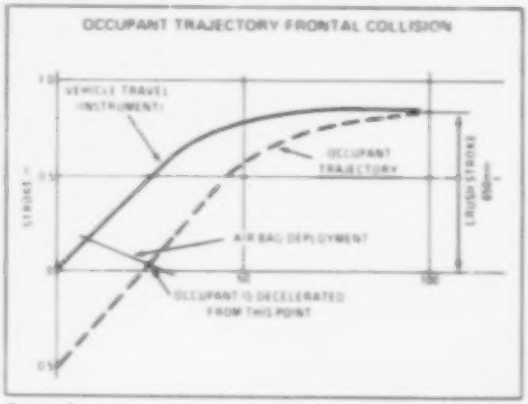


Figure 2

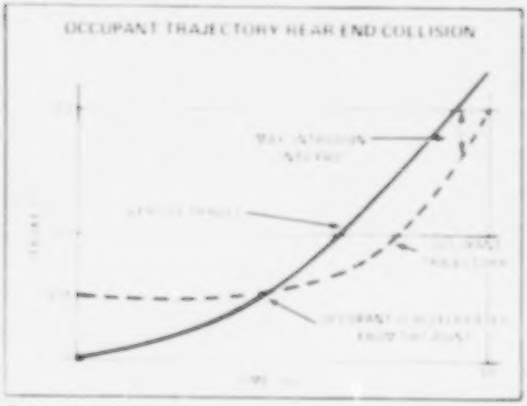


Figure 3

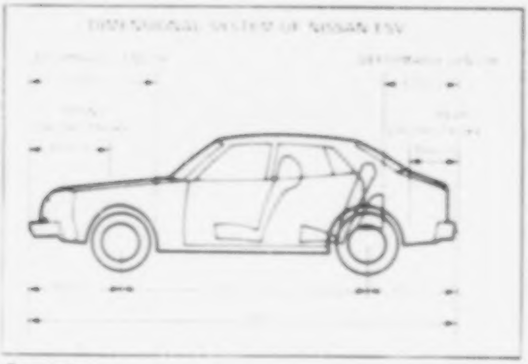


Figure 4

Figure 5

We have made efforts to produce the Nissan ESV as a small, attractive car within the above dimensions. The outer configuration has received consideration with a view to meeting the visibility specifications. The hood is low enough to fulfill the requirement of 6° downward visibility, and comply with pedestrian safety (Figure 6). The front end, occupant compartment, rear end, etc. are constructed for improvement in safety, and the inner configuration is designed with a view to occupant protection and practicality, in order to provide the occupants with a safe feeling.

Figure 6

ACCIDENT AVOIDANCE

Handling And Stability

We have adopted a rack-and-pinion type steering system, designed not to impede the crush stroke on the frontal body in a collision. We have also improved the installation of the steering system as a whole. This has also resulted in improved steering characteristics, such as quick responses.

Furthermore, for occupant protection in a collision, we have developed an anti-reaction shaft (Figure 7). This device prevents the steering column from thrusting into the passenger compartment, as the intermediate shaft, located between the steering column and the steering gear, retracts when loaded to a pre-determined degree.

As a result of our research, we have designed a rack-and-pinion type steering system as mentioned

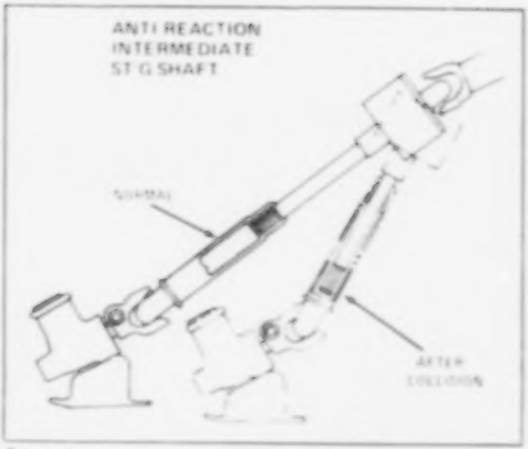


Figure 7

above, a strut-coil type front suspension (Figure 8) and a semi-trailing independent type rear suspension (Figure 9). The brake system is composed of a disc-type front brake, a drum-type rear brake with an aluminum cooling fin, a large-sized vacuum booster and skid control.

The braking, steering and handling of ESV prototypes tested to date have met the ESV specifications. In addition to the compliance tests, we have performed ESV practicality determination tests and durability tests by unit and by actual vehicle, and have made an integrated evaluation of the related parts.

For our ESV tires, we have developed a double wall type safety tire which not only meets the specifications but assures safe driving even after punctured (Figure 10). This has brought about the following results:

- Prevention of accidents resulting from a puncture when driving at a high speed.
- Freedom from danger of changing tires on a highway at night or in inclement weather.
- Elimination of the spare tire housing in order to secure the body crush stroke.

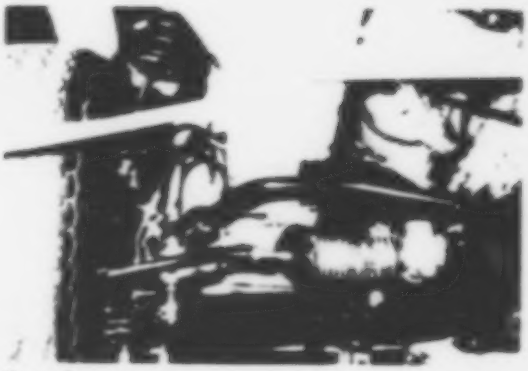


Figure 8

Figure 9

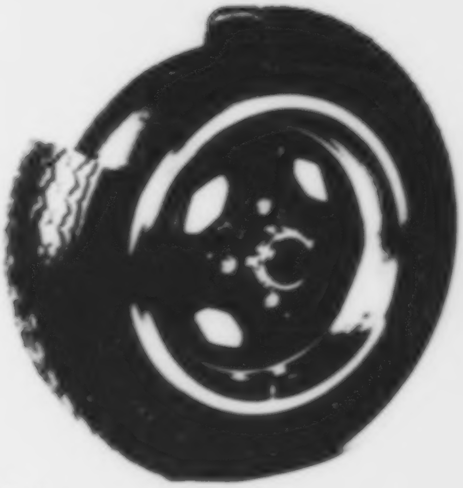


Figure 10

We have ensured that even after punctured, this tire enables the vehicle to travel at a speed of 80 km/h for 500 km without trouble.

As a result of testing a vehicle having a punctured tire, steering, braking and operating performance have proved to be sufficiently practical (Figures 11, 12, 13, 14, & 15). As there is apprehension that the driver may be unaware of the punctured tire, we have developed an accurate air pressure warning device. As is described later, unusual air pressure is indicated on a central warning device installed on the instrument panel.

Visibility

We have put forth effort to secure maximum visibility for our ESV, with consideration given to compliance with the requirements of the occupant protection system. In addition, we have studied the performance, arrangement, size and illumination of lighting systems and have made efforts to provide safety in all types of weather and in all environments.

STOPPING DISTANCE
FROM 95Km/h

NORMAL	44.2m
FRONT TIRE PUNCTURED	46.5m
REAR TIRE PUNCTURED	43.7m
ALL TIRES PUNCTURED	43.9m

Figure 11

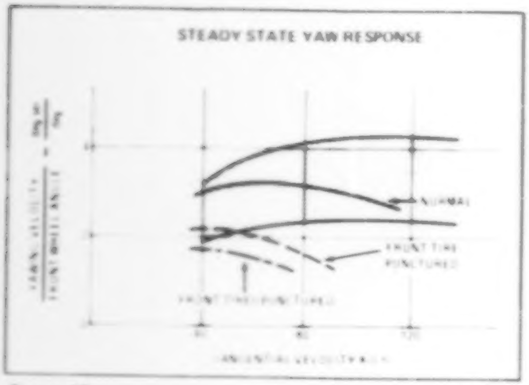


Figure 12

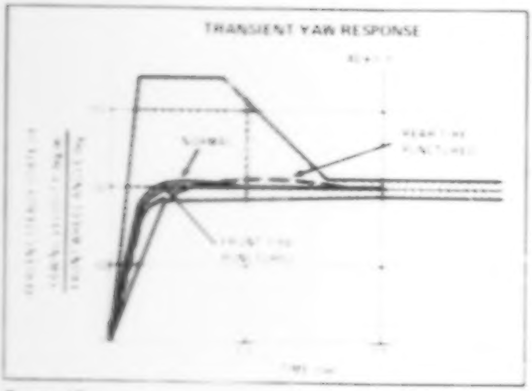


Figure 13

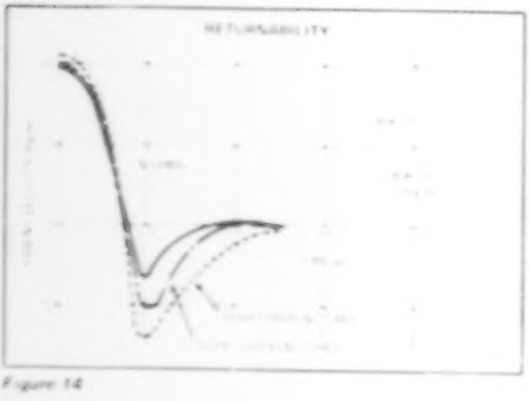


Figure 14

By adoption of a small-sized periscope provided with two concave mirrors, the rear visibility covers adequate range for ensuring safety (Figure 16). There are, however, some problems yet to be solved. Distortion of the image on the mirror, limited allowable eye range, difficulty in manufacturing this particular device, etc. are yet to be perfected and require evaluation and further study.

HANDLING AND OVERTURNING IMMUNITY TEST

[illegible]	NORMAL	FRONT TIRE PUNCTURED	REAR TIRE PUNCTURED
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]

Figure 15

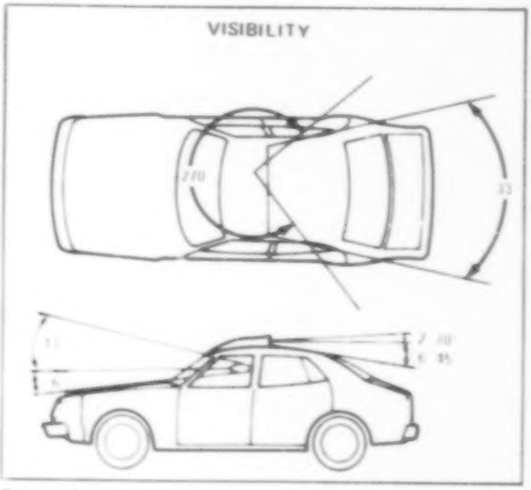


Figure 16

Additionally, a bright, 3-beam type headlamp (Figure 17), which is controlled by the car speed, has been adopted to improve the lighting pattern and visibility. The rear signal light is switched from daytime to night-time beams. The trail lamp unit adopted is devised so that a green lamp is activated during acceleration or ordinary driving, an amber lamp during deceleration, and a stop lamp remains on after stopped in order to allow a car following to immediately recognize the state of the ESV (Figure 18).

Further, our ESV is equipped with a central warning device. When trouble occurs with any item

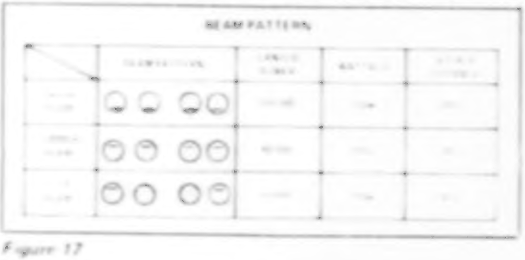

BEAM PATTERN

Figure 17

Figure 18

related to safety maintenance, such as shortage of air pressure in tires, breakage of lamp wiring, malfunctioning of the brake system or defective door lock, this device will reflect on the instrument panel the image of the spot out of order and will sound a buzzer. This has made it possible to diagnose problems while driving the vehicle.

OCCUPANT CRASH PROTECTION

Development Of System E_1

1. Crashworthiness

The body size has been limited to the maximum allowable extent, and, for a reduction in weight, highly-efficient, energy-absorbing members have been installed on all sides of the passenger compartment. For the purpose of withstanding a large reaction force during a collision, the compartment has been comparatively strengthened.

As a result of systematic analyses by computer simulations and tests with many modified vehicles, we have found:

- a. That the absorption of kinetic energy is not only accomplished by the frontal body structure, but, also is considerably shared by the engine intrusion into the dashboard and intrusion into the floor, and by the deformation of the power train system, and
- b. That accordingly, these factors are involved in the vehicle deceleration characteristic and peak G is produced to serve as a means to prevent the catastrophic deformation of the whole vehicle (Figure 19).

With System E_1, we have developed a method of appropriate distribution of vehicle energy absorption based on these analyses, and we have also developed a body construction which is a practical, light sheet metal panel structure, and which has an adequate energy absorption performance. As a consequence, crashworthiness in frontal collisions has almost attained target performance. In rear-end collisions, the monocoque construction has displayed nearly ideal crashworthiness.

2. Points in body construction (Figure 20)
   - a. Front member

     Energy is absorbed by crushing deformation of four upper and lower members.
   - b. Engine firewall (Figure 21)

     In order for the engine to recede smoothly in a frontal collision, the engine firewall is divided into three portions. Portions "A" and "B" are constructed to be free from deformation.
   - c. Floor (Figure 22)

     The tunnel and sill units are strengthened with high-tensile steel reinforcements, in order to adequately withstand compression and bending. The portions are joined by seam welding to increase strength.
   - d. Front and rear doors (Figure 23)

     Strong members withstanding, the front and rear load are installed inside the upper part of the door panel, to prevent deformation of the passenger compartment in a frontal barrier

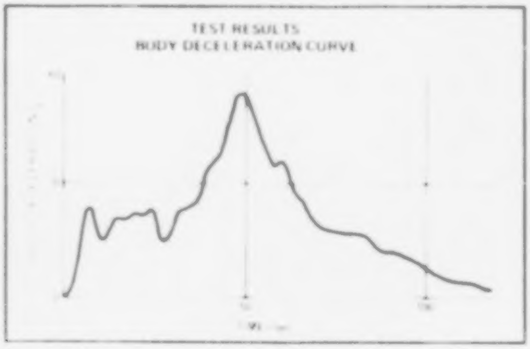


Figure 19

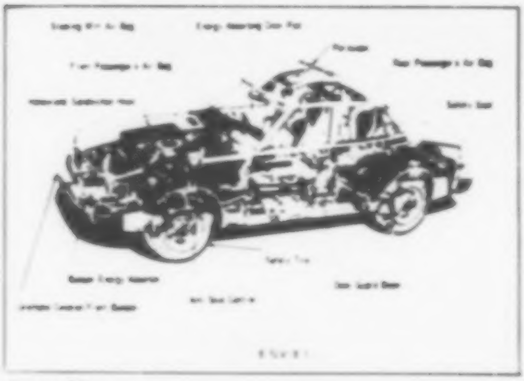


Figure 20

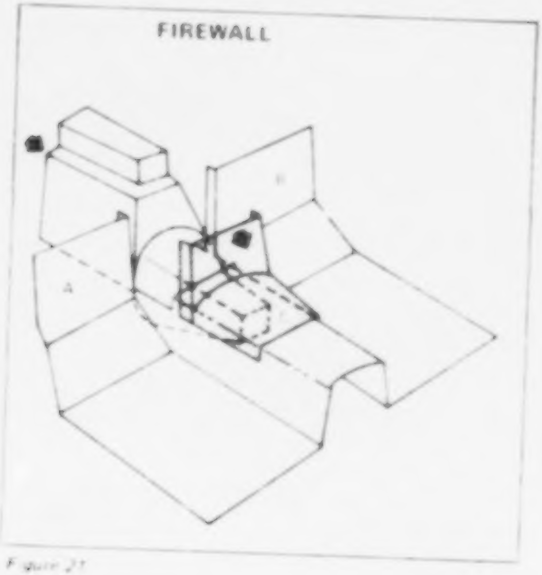

FIREWALL
A
B

Figure 21

FLOOR CROSS SECTION

Figure 22

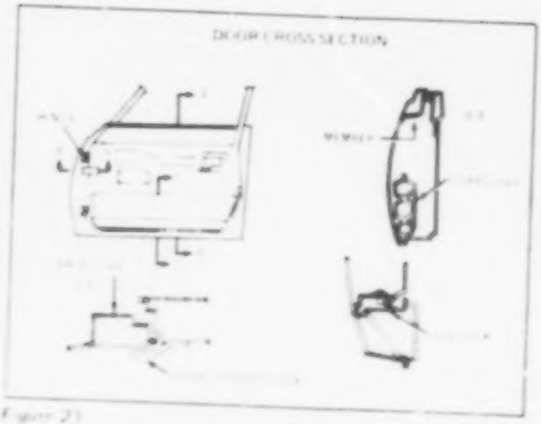

DOOR CROSS SECTION
MEMBER

Figure 23

collision. In addition, a stopper plate as a fail-safe device is installed to keep the door in place even if the hinge is broken. The possibility that the struck car in a side collision thrusts itself into the compartment is reduced to a minimum with the utilization of a large guard bar made of high tensile steel, and with a sublock.

e. Rear member (Figure 24)

The trunk-floor member of a double-hat section, and the box-type section of the rear fender unit are designed to absorb energy during a rear-end collision.

f. Bumpers (Figure 25)

Both front and rear bumpers are equipped with hydraulic shock absorbers. These shock absorbers have shock absorbability, which prevents the running system from sustaining serious damage in a light collision with a fixed barrier at a speed of 15 km/h. In addition, they have the function of absorbing partial energy in a high-speed crash.

g. Urethane injection and foaming (Figure 26)

In order to secure the occupant survival space in case of a roll-over accident, the roof of

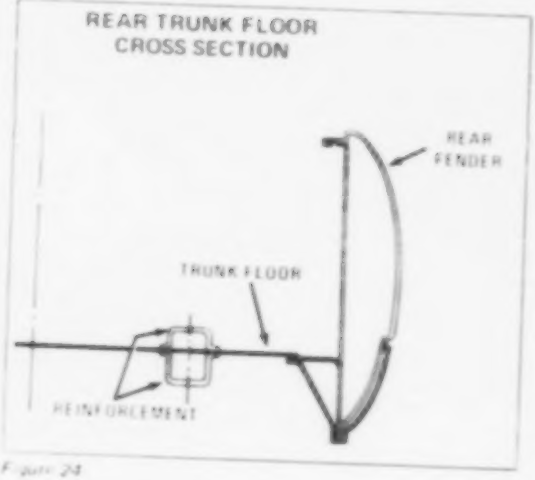

REAR TRUNK FLOOR
CROSS SECTION
REAR
FENDER
TRUNK FLOOR
REINFORCEMENT

Figure 24

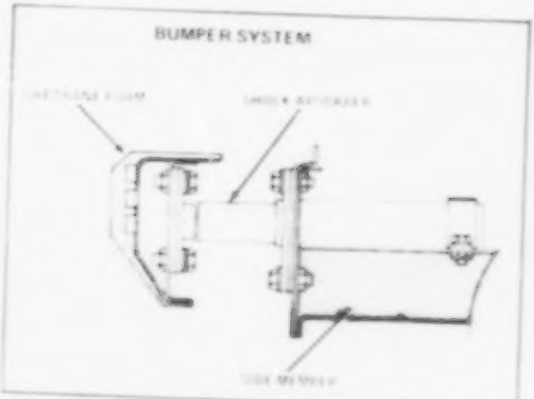

BUMPER SYSTEM
URETHANE FOAM
SHOCK ABSORBER
SIDE MEMBER

Figure 25

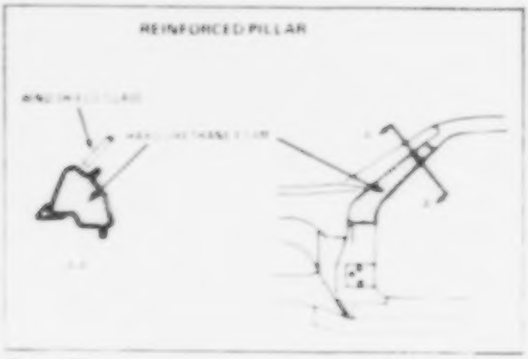


Figure 26

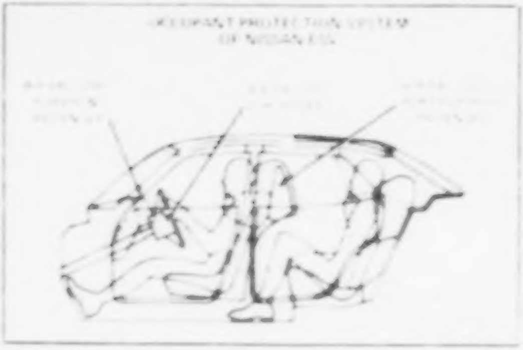


Figure 27

A-pillar is strengthened, and the pillar has had hard urethane injected for foaming and has had bending strength increased three-fold.

b. Reduction in weight

For lighter weight, the front fenders, trunk lid, and cowl grille are made of aluminum. For reinforcements in the compartment, highly tensile steel sheets (tensile strength of 50 kg/mm² to 80 kg/mm²) are utilized.

(3) E. Occupant protection system

a. Occupant protection during frontal collisions (Figure 27)

As already stated, we have developed a fast-deployable air bag so that deceleration may be produced in the occupant within 25 milliseconds after the collision. The air bag is a pressure-controlled deformation-speed type air bag.

For detection of a collision, a G-sensor is adopted. This sensor, when installed on the radiator support member, can easily perceive a frontal collision within 7 milliseconds after the initiation of a crash.

Aside from this sensor, we have developed a radar sensor which forecasts a collision before it takes place. We have been developing pulsed Doppler Method as well as Multi Frequency Doppler Method as a radar system, which detects relative distance and relative speed. Protection of the occupant's waist and knees are performed by a lap belt and knee pad. Without the lap belt, the injury criteria provided in the ESV specifications can be complied with, but the utilization of the lap belt approves the recurrence of occupant motion. While the air bag is deploying, air pressure in the compartment is increased by 0.14 kg/cm², and noise level thereon as high as 173 dB. So, the rear window is destroyed for the purpose of reducing air pressure in the compartment. Of all seats, the driver's seat is most difficult to protect, and it is protected by an 80-liter air bag mounted on the steering wheel (Figure 28), an energy-absorbing type steering column, a knee pad, and a 3-point type seat belt. The front passenger seat is protected by a 200-liter air bag housed in the instrument panel, a lap belt, and a knee pad.

The rear seat occupant is protected by a 150-liter air bag installed in the front seat back, a lap belt, and a knee protector attached to the front seat. The front seat, which undergoes the reaction force of the air bag for the rear seat occupant, is a rigid structure equipped with a frame made of high-tensile steel.

The front seat (Figure 29) is a separate sliding type. Therefore, the seat slide and the floor are reinforced in comparison with those of conventional vehicles.

b. Occupant protection during rear-end collisions

Occupant protection during rear-end collisions is provided by a high-back seat with an attached headrest (Figure 30), and by energy-absorbing material installed in the seat back. The front seat is sufficiently strengthened so as not to fall

Figure 28

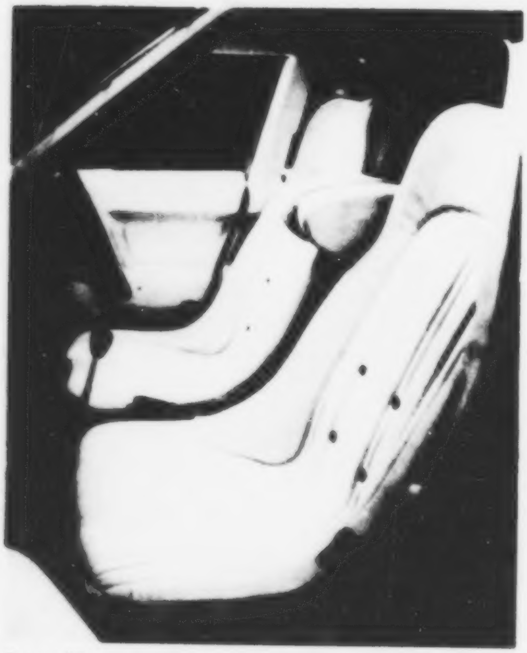


Figure 29

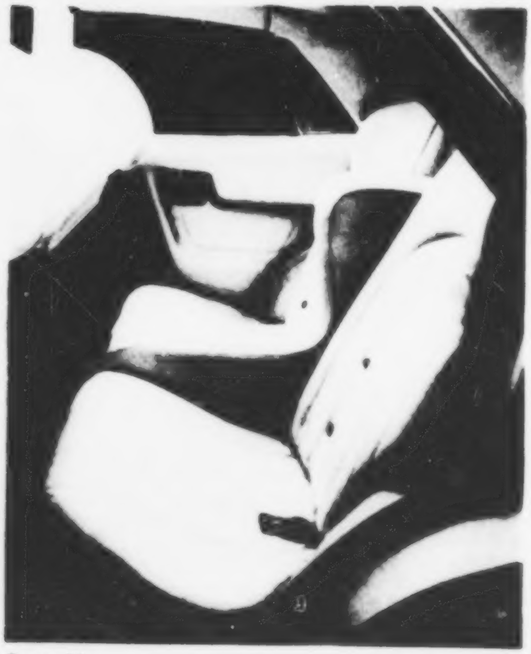


Figure 30

rearward during rear-end collisions and is prevented from having an adverse effect on the occupant of the rear seat

c. Occupant protection during side collisions

During a side collision, the sides of the compartment are protected by A, B and C-pillar pads and door pads. Each pad is about 50 mm thick. The door glass is composed of laminated, chemically-tempered, energy-absorbing glass of recent development, for the purpose of providing further occupant protection. Moreover, each seat is provided with a shoulder protector, to prevent the occupant from colliding with the occupant in the next seat.

d. Others

For occupant protection during a roll-over accident, 30 mm padding is installed in the compartment ceiling.

Results Of Tests

During all frontal, rear-end, side-collision tests (Figures 31, 32 & 33) and roll-over tests performed with ESV prototypes, we have ensured that occupant survival space in the compartment and all occupant injury values have satisfied the target criteria.

Regarding the injury values of the driver's seat occupant obtained in 80 km/h frontal barrier collision tests, some have not yet met all criteria. Therefore, we are proceeding with ESV development for further improvements in the occupant protection system. The typical G waves of the occupants of the driver's, front passenger's and rear passenger's seats during 80 km/h frontal barrier collisions are as shown in Figures 34 and 35.

Concerning frontal collisions with a rigid pole, the Nissan ESV is designed to withstand a collision speed of 65 km/h. As to the test condition, it is clear that when the strength or weight of the cross member installed in the body front is increased, the guaranteed speed is increased proportionately. This is not a matter of great interest.

Regarding frontal car-to-car collisions, when our ESV was caused to collide with a large-sized vehicle of 1,800 kg (4,000 pounds) at a relative speed of about 90 km/h, (Figure 36) both cars sustained

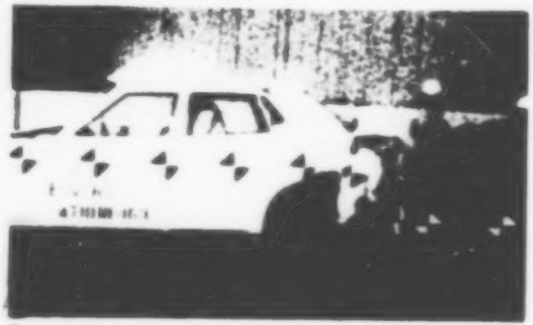


Figure 31

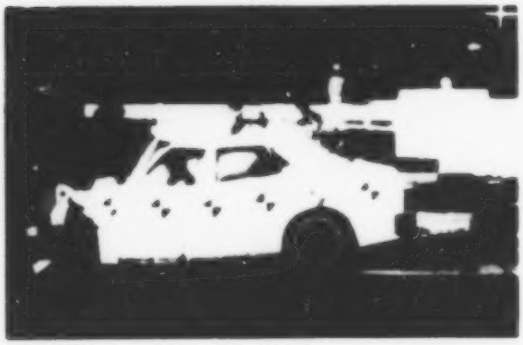


Figure 32

Figure 33

nearly the same degree of crush, and it was ensured that the occupant safety was also secured during collisions. At higher speeds, the Nissan ESV sustained a more serious crush, greatly affected by the strength of the large vehicle's frame. In consideration of high-speed collisions, large-sized cars should have their front units made softer.

Development Of System E_2

As stated earlier, System E_1, based on the use of air bags, is nearly the same in basic construction as vehicles on the market, and is intended for the possibility of mass production. Conversely, System E_2 is a vehicle system brought into being only as a result of accumulated studies of necessary units, and is intended to comply with the injury criteria in 80 km/h frontal barrier collisions by utilization of the seat belt.

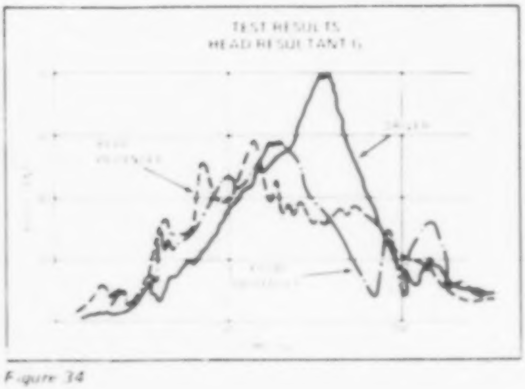


Figure 34

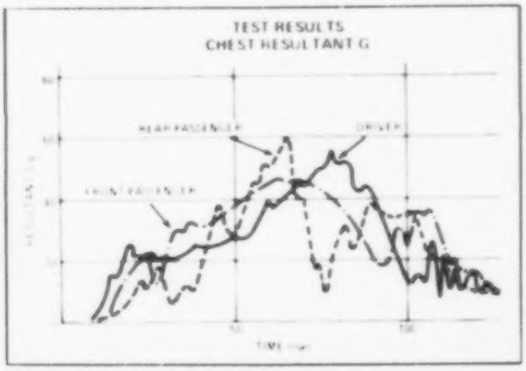


Figure 35

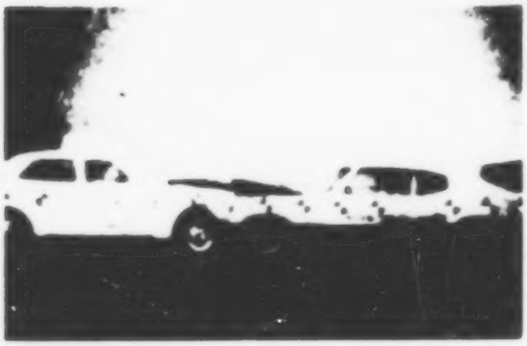


Figure 36

1. Front-end structure of System E_2

   For improvement in energy absorbability, the front end of System E_2 is of a broach-type construction, which utilizes the expansion of pipe (Figure 37). The broach characteristic obtained is similar to a square wave (Figure 38), and the stroke

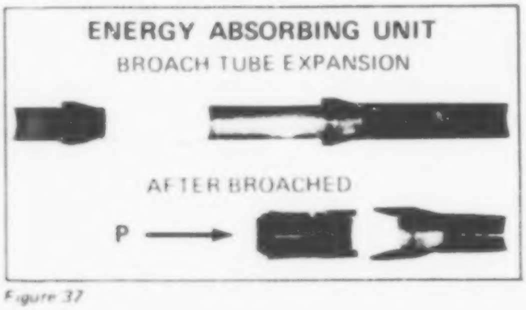


Figure 37

efficiency is raised to 70% by having the broach unit constructed as a multi-stage type. Further, by changing the diameter, thickness, expansion ratio and die angle of the broach pipe, energy absorbability can be easily controlled and matched with the seat belt system.

During 80 km/h frontal fixed-barrier collision tests, the deceleration waves of vehicles with a broach construction have irregularity. Such irregularity, appearing immediately after the crash as revealed by a simulation to be referred to later, results from the vibration of the body assembly and rear suspension system. It is therefore difficult to eliminate the first irregularity on real vehicles. The average load on the G-S curve, however, corresponds to the designed broach load (Figure 39).

The broach load is designed so that about 70% of all kinetic energy of the vehicle is absorbed by the front-end structure. This rate has been determined as a result of computation made in view of the fact that the effective mass of the front-end structure undergoes change with the passing of time after crash begins. The front-end structure of System E_2 is basically composed of a 2-stage upper broach and a 3-stage lower broach, with both broaches connected by the hood ridge panel which forms the engine compartment (Figure 40).

The front bumper and the hydraulic shock absorber at the end of the lower broach are connected by means of a pin-type device to withstand an angular collision.

2. Occupant compartment structure (Figure 41)

When the front end is a broach type, the occupant compartment requires partially greater intensity than in the case of System E_1, in order to withstand concentrated heavy load from the front end.

The arrow marks of different sizes in Figure 42 show the magnitudes of force transmitted to the respective units, computed by a simulation of 10 mass and 19 non-linear spring model.

The process of vehicle deformation in a collision is shown in Figure 43. Additionally, body static compression tests have been conducted for improving the strength of the passenger compartment (Figure 44).

3. Points of passenger compartment structure

Parts different from those of System E_1 are as follows:

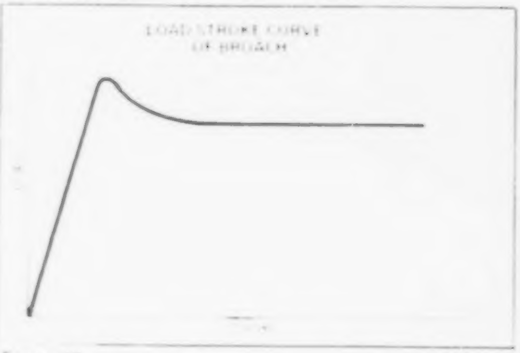


Figure 38

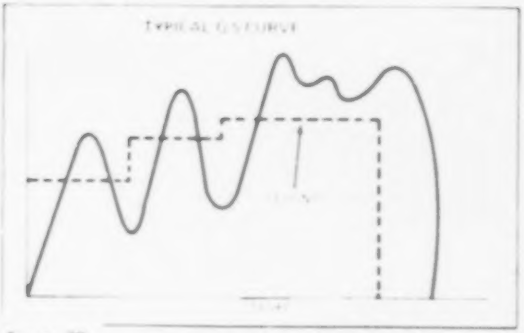


Figure 39

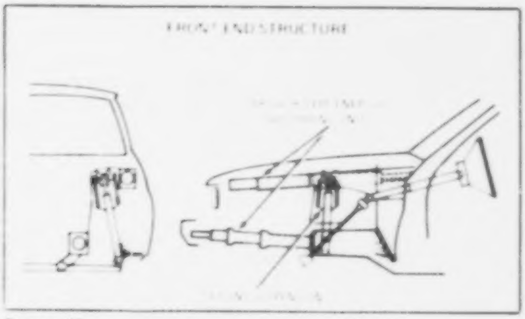


Figure 40

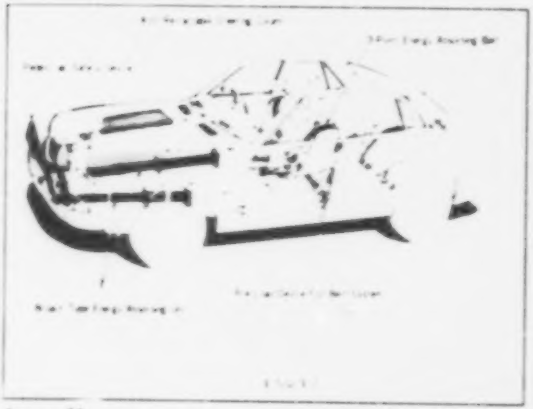


Figure 41

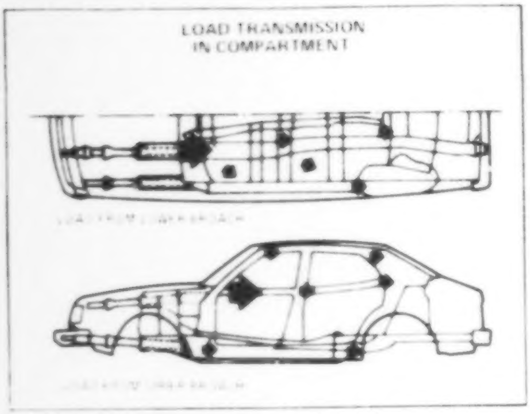

LOAD TRANSMISSION
IN COMPARTMENT

Figure 42

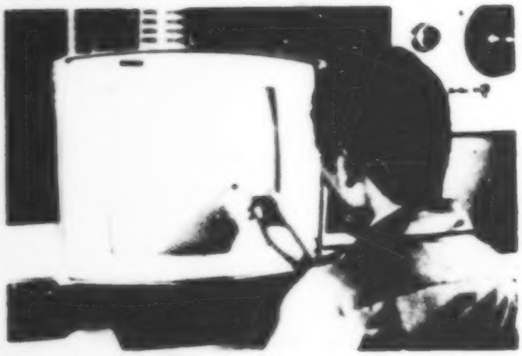


Figure 43

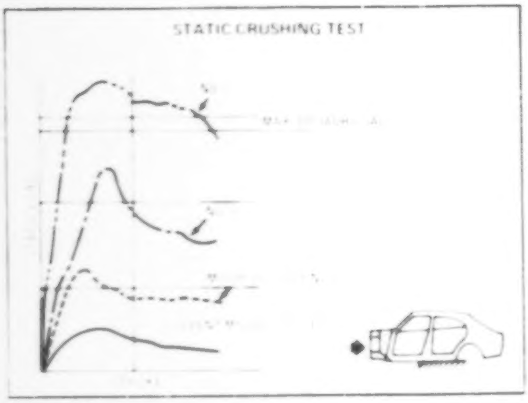


Figure 44

- A-pillar

  The A-pillar has a built-in baffle plate in order to adequately withstand the upper broach load and the door's inertia (Figure 45). The shape of the baffle plate is designed to prevent the door from coming off.

- Door

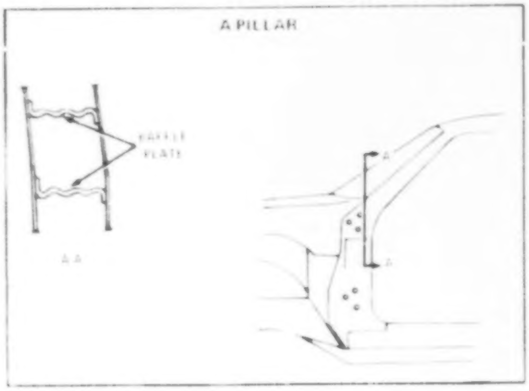

A PILLAR
BAFFLE PLATE
A-A
A
A

Figure 45

  The door member above the belt line has a polygon box-type section to resist crush load (Figure 46).

- Floor

  The front side member and the floor member are also of polygon design, with more intensity. A gusset is installed so that the lower broach load will be surely transmitted to the tunnel reinforcements (Figure 47).

4. E[illegible] Occupant protection system

Previously, it was believed that a seat-belt system effective in an 80 km/h collision would be difficult to develop. The reasons were:

a. In such a collision, the occupant's head is pushed further forward than when it is restrained by an air bag, and more occupant survival space is necessary.

b. To restrain the head is not easy, because its rotation involves greater centrifugal force.

We attempted to solve these problems by producing a maximum effect of "ride down," with

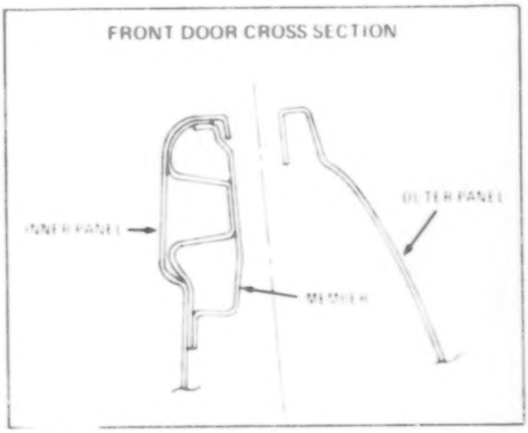

FRONT DOOR CROSS SECTION
INNER PANEL
OUTER PANEL
MEMBER

Figure 46

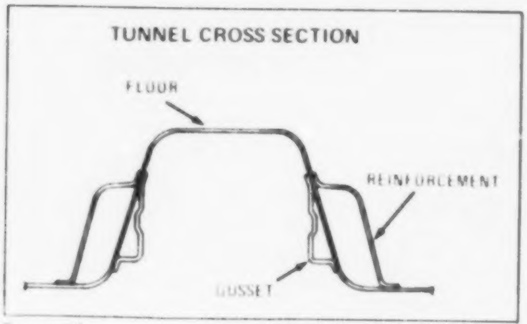


Figure 47

improvement in the belt characteristics and with adjustment to the deceleration G wave of the vehicle. As to the relation between the vehicle deceleration wave and occupant deceleration, we utilized a simple dynamic model, in an attempt to find optimum vehicle characteristics.

In designing the belt system, we performed a computer simulation by a two-dimensional model of 5 masses and 7 freedom degrees, for the optimum combination of belt tension, belt anchor installation position, knee pad characteristics and vehicle deceleration wave. As a result, it was found that the ESV injury criteria can be satisfied by using a 3-point belt (Figures 48 & 49).

The actual belt system is devised as follows (Figure 50):

- The shoulder-belt anchors are installed immediately behind the shoulders. Particularly, all anchors for the front seats are installed in the seats.
- Force limiting type energy-absorbing belts are used in the shoulder belt and the inside lap belt.
- For the belt unit material, high-tensile tetron thread is used to quicken the onset rate of the load.

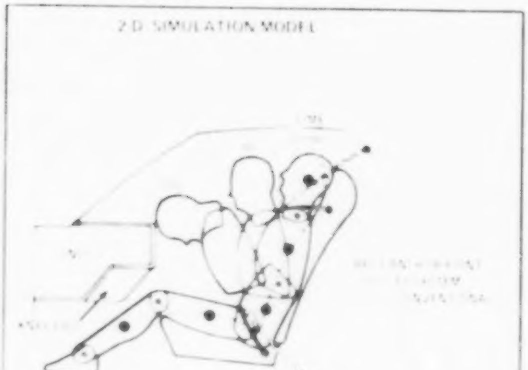


Figure 48

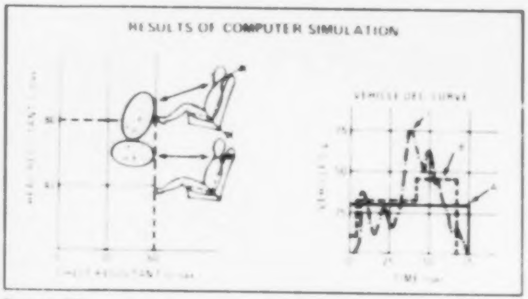


Figure 49

With this system, the front passenger seat and the rear passenger seats have achieved values to satisfy the criteria (Figure 51). The driver's seat was, however, short of the restraint stroke. We have further developed a pre-loaded type seat-belt device (Figure 52) and an auto-retractable collapsible steering column (Figure 53).

As a load generator for these devices, we have adopted a high-pressure gas type unit which is

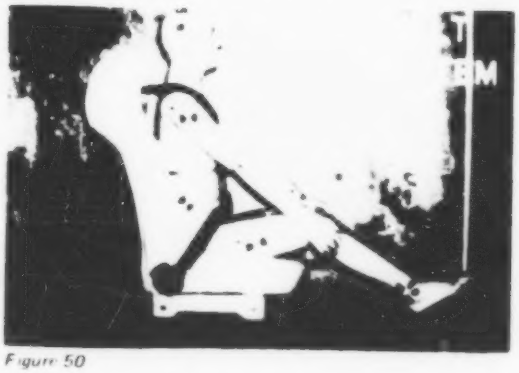


Figure 50

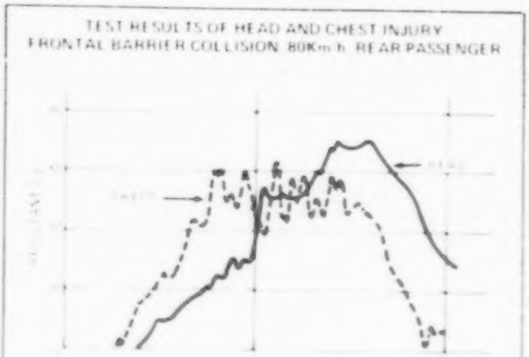


Figure 51

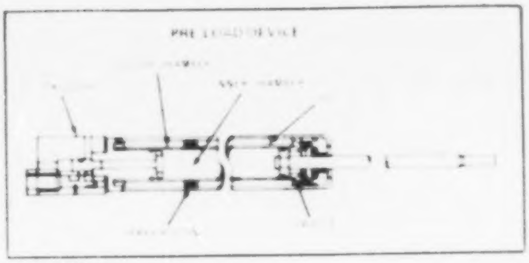


Figure 52

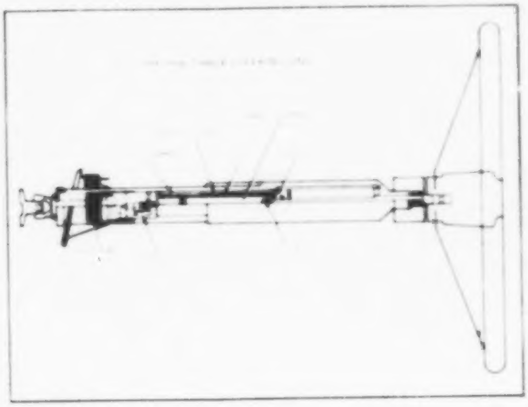


Figure 53

triggered by a collision-detecting sensor.

Generally, when the belt slack is reduced, it becomes possible to decrease the occupant stroke with occupant deceleration kept low. The reason is that that part of the occupant's kinetic energy, which is absorbed by "ride down," is rapidly increased. Figure 54 indicates this relation with ride down ratio (η) obtained by utilizing a simple dynamic model. From Figure 55, the effectiveness of preload can be fully understood. With the development of these new devices, it has become possible for even the driver's seat to get values complying with ESV injury criteria (Figure 56).

PEDESTRIAN SAFETY

Even a little improvement in pedestrian safety, affected by the vehicle itself, should be considered of great value. From this standpoint, we have attempted to develop an effective pedestrian safety device, as referred to in the following.

A pedestrian sent flying by a vehicle will first sustain a leg bone fracture in a leg-to-bumper impact, or a first collision. In a 15 km/h collision, the pedestrian can be checked from sustaining such a fracture when the bumper surface is covered with thick urethane, which has been made hollow for improvement in energy absorbability (Figure 57). This device will be effective in preventing a pedestrian from receiving a multiple fracture, even in a collision at more than 15 km/h.

The pedestrian sent flying will suffer a collision with the front end structure, or a second collision. In

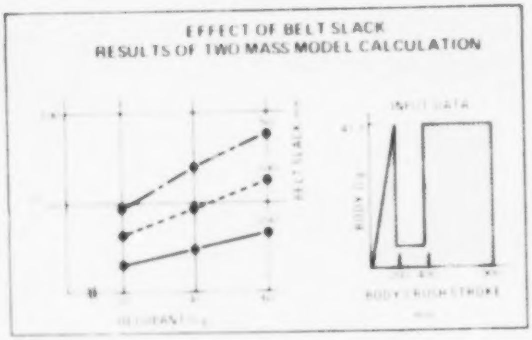


Figure 54

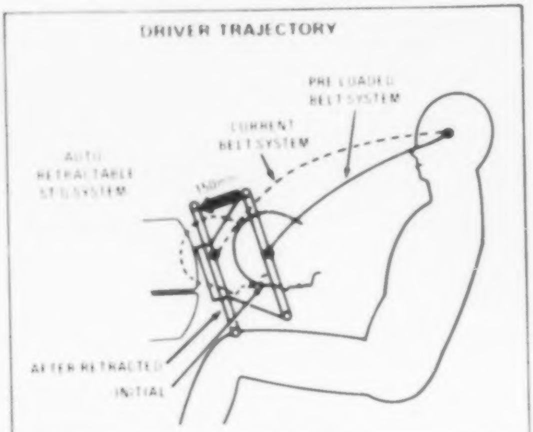


Figure 55

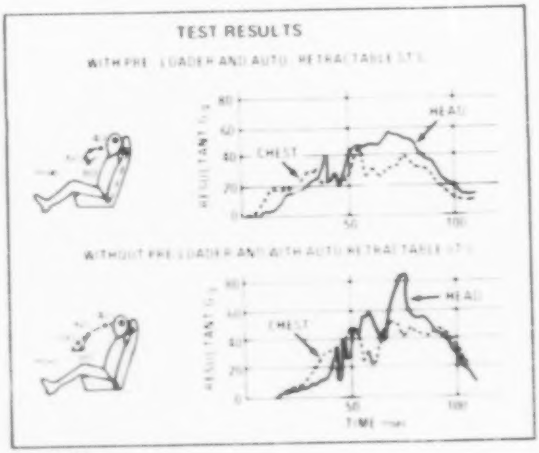


Figure 56

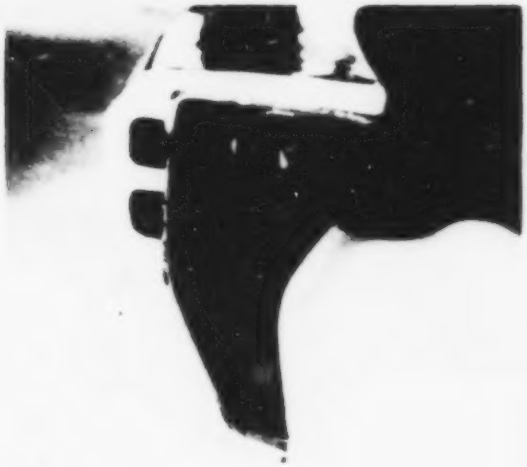


Figure 57

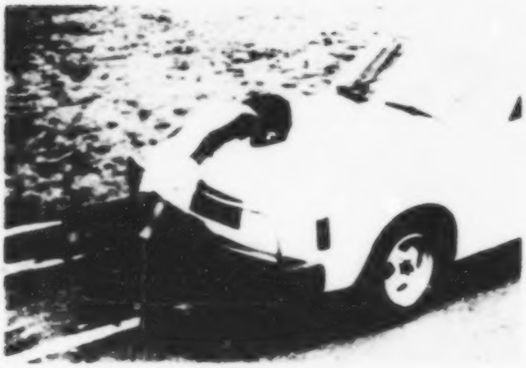


Figure 58

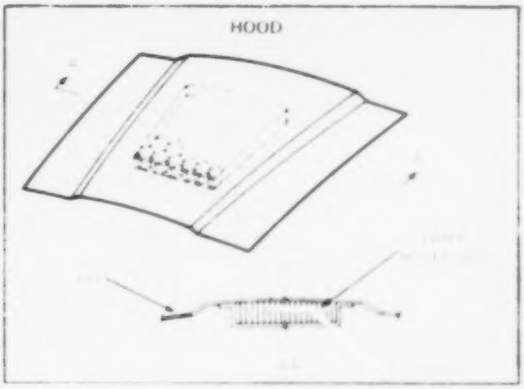


Figure 59

collisions at more than 15 km/h with a small car, the pedestrian will be sent flying against the front glass, sometimes receiving an injury or even meeting death. To prevent this, a pedestrian safety device is installed to control the pedestrian's trajectory, so that he may collide with the hood, as close to the center as possible (Figure 58).

The hood has a built-in honeycomb for impact energy absorption (Figure 59). If the pedestrian should be thrown to the road surface, he would receive a serious injury. This pedestrian safety device is so designed that in some cases it may catch the pedestrian falling from the hood and prevent him from crashing directly to the road surface. In collision tests carried out at less than 30 km/h with an AM 50 percentile dummy, we have found that the foregoing purposes are met to some extent.

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART I

EVALUATION OF THE NISSAN ESV

YOSHIO SERISAWA, *Manager*
Safety Research Department
Nissan Motor Company, Ltd

As stated previously, Nissan Motor Co., Ltd. has experimentally achieved the ESV specifications. When seen as an automobile, however, the Nissan ESV has shortcomings, and its values complying with the criteria are those to be attained only by prototype vehicles manufactured with stringent control. Moreover, it is likely that with a revision of the many criteria the Nissan ESV would undergo a great change.

PROBLEMS IN OCCUPANT PROTECTION SYSTEM

1. The air bag developed for the E_2 vehicle is remarkably quick in deployment and outstanding in performance. On the other hand, the air bag has such problems that its deployment causes higher pressure and extremely greater noise level in the compartment. The chemicals used for combustion acceleration frequently cause abnormal combustion and some irregularities. Moreover, a maloperation of the air bag will have an adverse effect on the occupant, giving rise to serious trouble. Thus, concerning the air bag, there are many problems to be solved. At this writing, we cannot determine when the air bag for 80 km/h collisions will be commercialized.
2. With the E_2 vehicle, the belt system requires a pre-load device and a retractable steering column unit for occupant protection in collisions at speeds as high as 80 km/h. In the experimental stages these units have been developed successfully. Now, however, we have many problems to be solved before they are offered for practical use.

RELIABILITY OF ESV MANUFACTURING

The Nissan ESV has been manufactured with the body intended to have maximum energy absorbability and with quality control conducted on every inch for intensity of any spot-welded joint. Concerning air bags, which are required to be fast deployable, sled tests on related units have been conducted to select outstanding samples for use on vehicles to be tested. To secure reliability in such a manner is considered impossible during mass production.

PRODUCTIVITY

The complexity of the body construction proper presents problems which are discussed in the following subparagraphs. Even with considerable changes in equipment and facilities, the ESV cannot be brought to the production stage:

1. For ESV construction work, including spot, seam and arc welding involving a lot of time and labor, a great increase in manhours and a great many additional pieces of equipment are required.
2. High tensile steel, aluminum, and FRP are utilized in large quantities. As a consequence, there are many technical problems yet to be solved in this respect.

WEIGHT AND COST

1. The Nissan ESV weighs a total of approximately 1,250 kg by curb weight. In comparison with the Nissan Datsun 610, the occupant compartment which is approximately of the same dimensions, is nearly 200 kg heavier in curb weight, about 95 kg heavier in the weight of the body construction proper (Table 1).

   The weight increase in the body construction proper, classified and computed by Nissan ESV specifications, is shown in Table 2. The reason why the total of each increment shown is more than 95 kg is that the reinforcement of the occupant com-

COMPARISON OF WEIGHT (kg)

[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
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[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]

Table 1

WEIGHT INCREASE OF BODY CONSTRUCTION	
	INCREMENT
80Km/h FRONTAL COLLISION	+95 Kg
65Km/h REAR END COLLISION	+25 Kg
50Km/h SIDE IMPACT	+50 Kg
ROLL OVER TEST	+25 Kg

Table 2

partment is functioning to some aspect. This is a noted feature of the Nissan ESV.

2. The production cost of the Nissan ESV is approximately 50% greater than that of the Datsun 610. This rate is based on the estimation that a monthly output of 10,000 ESVs is accomplished with the existing equipment and current techniques, regardless of production problems referred to in Table 3.

PRACTICALITY PROBLEMS

The Nissan ESV is essentially the same as the Datsun 610 regarding the dimensions of the passenger compartment structure. Practically speaking, there are the following problems:

1. In comparison with the Datsun 610, the effective width of the passenger compartment in the Nissan ESV is about 100 mm less, and the height of the passenger compartment is about 30 mm less. This is due to the thick padding for occupant protection.
2. The trunk capacity is about 40% less than that of the Datsun 610. This is partly because the rear overhang is shortened to effect a reduction in total length, and partly because improvement is made in the safety of the fuel tank.
3. The acceleration performance (SS 400 m) is slower by about four seconds than that of the Datsun 610. This is a result of an increase in car weight.
4. The practical minimum turning diameter (wall to wall) is about 600 mm greater than that of the Datsun 610. This is because the front end is made longer. For this size of vehicle, easiness in handling is considerably sacrificed.

Table 3

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART 1

FUTURE SAFETY CONSIDERATIONS

YOSHIO SERISAWA, *Manager*
Safety Research Department
Nissan Motor Company, Ltd

1. In view of air pollution, traffic congestion and a shortage of natural resources, vehicles should be made smaller and lighter. The majority of vehicles used in Japan, at present, are those of the 700 kg to 1,000 kg class. It is considered appropriate that popular vehicles be limited to 1,000 kg (2,200 pounds).
2. In this size, practical cars worthy of 80 km/h frontal collisions should not be manufactured. Traffic accident statistics indicate that vehicles capable of 50 km/h frontal collisions are considered contributory to safety. Therefore, there is a matter of urgency to develop practical safety vehicles worthy of 50 km/h frontal collisions. Vehicles for 65 km/h frontal collisions are considered to be a target for future development.
3. At present, specifications provided in detail are not appropriate for manufacturing complying vehicles. With mutual effort, we should establish appropriate performance standards, including injury criteria.
4. We have challenged the development of a seat-belt-provided vehicle worthy of 80 km/h frontal collisions, and have evidenced considerable progress. Although belt-fitted vehicles for 80 km/h collisions cannot be realized in practical use, we wish to emphasize the value of seat belts.
5. Regarding the matter of aggressiveness in passenger car collisions, large-sized vehicles should possess a large stroke, thereby making reaction force smaller. In so doing, both cars in a collision will have equalized damage. We recommend that along with this idea, the barrier crashworthiness requirements for large-sized cars should be determined, taking into account those for small cars.
6. As to accident avoidance, whole system evaluation criteria, covering the driver's performance as well, will be required in the future. In addition, clarification of the relation between car performance test data and actual safety should be required.
7. To be sure, a prototype vehicle should be required to have overall safety performance, but its appropriate cost-benefits should also be considered. If all this is carefully analyzed, the overall increase in cost will be lowered so that we can produce and offer a safer vehicle at a lower price for all people.
8. Safety should not only be uniformly regulated, but also desired by the consumer. In this sense, higher and more expensive safety performance would be the result of consumer demand.
9. Development of safety vehicles is of course valuable. On the other hand, it is also very effective to control roads and drivers for reducing the traffic accidents. We expect that there will be international cooperation in this respect as well.
10. In addition to the fulfillment of requirements set for motor vehicles, we also have to solve the problems of safety, air pollution, traffic congestion and shortage of natural resources. It is by no means an easy task to give a rational and balanced answer to a universal proposition calling for "the greatest happiness of the greatest number for the longest period of time." This is, however, a goal worthy of us all.

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO PART 1

THE PRESENT DEVELOPMENT STATUS OF THE HONDA ESV

KIYOSHI KAWASHIMA, *General Director*
Honda Research and Development Company, Ltd

Through the ESV program currently being undertaken on a worldwide scale, Honda has made a number of achievements in pursuing the technical requirements needed for an ESV.

As you are well aware, an automobile must be designed with a concept of overall balance that contains numerous characteristics. In the 2000-pound class small ESV concept projected by Honda, chief emphasis was placed on the well-balanced combination of various safety requirements imposed on ESV as a future automobile, in addition to the small car abilities needed in city traffic. Consequently, Honda ESV requirements can be summarized as follows:

1. Unique concept of small automobiles.
2. Thorough study of the limit of accident avoidance performance of a small vehicle.
3. Thorough study of the limit of crashworthiness of small vehicle.
4. Information system to communicate between human drivers and machine (automobile).
5. Cost benefit balance with consideration of marketable products.
6. Anti-pollution measures.

Exhaust emissions, as a major social problem, is equally as important as the safety problem now being controlled by the Federal Motor Vehicle Standards, and various approaches have been and are being made to solve the pollution problem on a worldwide scale.

In the near future, in order to completely integrate these two themes in an automobile, and to demonstrate its effectiveness sufficiently, a study to seek those balances must be made.

Safety and pollution problems must be worked out side by side, like the right and left wheels of a vehicle. As a next step, ESVs must be developed with a higher concept for considering their feasibility as a marketable product and as a partial solution to these two parallel problems.

Concerning crashworthiness, which is a very difficult problem for a small ESV, we established a most rational goal and have made our best effort in reaching it without destroying the overall balance that an automobile must have. The specific goal was disclosed at the 3rd International ESV Conference and the Honda ESV being introduced at this conference has almost achieved that goal.

In spite of our effort, the weight of the Honda ESV has exceeded the initial design weight of 1,500 pounds and will be around 1,900 pounds when completed.

Fulfilling all of the ESV requirements results in weight increase. When we try to balance the various phases in achieving the feasibility of our ESV as a future marketable product, we believe that the weight can be somewhat reduced.

While improving the handling and stability, we have also carefully considered the overall balance of our ESV so that the riding comfort will not be sacrificed.

Although there was a weight disadvantage because of the circumstances mentioned above, a hydraulic control was adopted on the suspension system as a full-power system to obtain those three major characteristics (small car abilities, handling and stability, and riding comfort) simultaneously.

The Honda ESV concept briefly explained above was explored specifically in a number of subsystems, and the ESV displayed at this conference is the integration of those subsystems. We are considering this as a reply to the ESV proposition and believe that our ESV, coupled with the embodiment of the CVCC system, can contribute to society in the transportation system of the near future.

In today's presentation, we would like to discuss accident avoidance performance and our evaluation of its safety aspect, since we believe that our approach has a number of advantages.

I would like to introduce Mr. Hayano, our staff engineer, who will discuss that phase of our ESV.

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THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO

PART 1

STRUCTURES AND ACCIDENT AVOIDANCE

HIROSHI HAYANO, *ESV Project Manager*
Honda Research and Development Co. Ltd

Firstly, I would like to describe and report on the outline of the Honda ESV, with emphasis placed on those items that differ from what were announced previously.

BODY STRUCTURE AND ARRANGEMENT OF ENERGY ABSORPTION UNITS

The body structure is a two-door type, with the rear seat serving as an emergency exit. From the start, comparison was made for superiority of crashworthiness between small car models with two doors and with four doors. However, each has its merits and demerits in performance, so that with consideration given to weight and cost, we elected to adopt a two-door type. To assure the post-crash escape of rear occupants, the emergency exit door opens with a single action, operable from both inside and outside the vehicle.

Highly Efficient EAU. Because of weight, we rejected adoption of the crash-energy absorption system by hydraulic pressure and have been studying an absorption system utilizing mainly plastic deformation of steel sheets. The system, as announced previously, consists of boxlike units made of four corrugated sheets, with two units placed in each end of the body. However, to make it function more efficiently, the system is now improved at the front by installation of four, instead of two, units so as to reduce the load to be received by each. As a result of this improvement, the EAU's will collapse more evenly and thereby further enhance the energy-absorption efficiency of the vehicle as a whole.

Impactless Steering Column. As part of the occupant-protection system, passive seatbelts are installed on the front seat. In order to improve a combined efficiency of the webbing and inertia retractor, in addition to full utilization of the former, a system is employed which, in the event of head-on crash, the steering wheel will move resistlessly and axially downward toward the instrument panel. The system is designed to assure an unobstructed space in front of the driver by unlocking the steering column to enable it to shorten approximately 150 mm by action of a sensor, which detects deceleration and actuates by inertia mass. In addition, the steering column is hydraulically and steplessly adjustable, by means of incorporated adjustment valves, in both vertical and longitudinal positions.

ACCIDENT AVOIDANCE ABILITY OF HIGHER LEVEL

In making the best use of their size, it is important for small cars to enhance the accident avoidance ability by efficient utilization of the maneuverability peculiar to them. However, if response and damping of the vehicle are raised, its gain becomes so high that it is likely to have an adverse effect by leading to an error in reactive operations by man. This is a problem which depends basically on the size of the vehicle's wheelbase and tread and relates to the stability theory seeking to obtain proper values for the gain characteristics. From the results of testing and computer simulation, we have determined the overall final dimensions as well as specifications for the suspension block of the exhibit car.

Of course, it was confirmed that the resultant improvement in the handling and stability enabled it to meet the specifications for those required of ESV; nevertheless, in an effort to develop a more practical ability for the car to avoid accidents, we will continue to improve the components of the suspension and steering systems. The effect, among others, which was brought forth by improvement in the tire characteristics and in the overall efficiency of the brakes, as well as by development of the skid control system is quite noticeable, with the result meeting our expectation. Particularly for the tire, it is believed that not only its performance was improved, but also it was enhanced in all accounts, including warning and prevention against irreparable damage, with emphasis placed on reaction by the driver to any condition requiring his attention.

The above is the gist of my report on the Honda ESV. However, I wish to take this opportunity to describe briefly our plans for the future development with respect to the accident avoidance ability.

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As a rule, the level of quality of a vehicle's handling and stability should generally correspond with the degree of safety provided by the vehicle when it is running. However, the system enabling the vehicle to run safely is quite sophisticated; it could be said that it is governed by a delicate man-machine-environment balance. Consequently, it is extremely difficult, if not impossible, to establish each of the criteria for evaluation of the vehicle's handling and stability corresponding to such a system. In actuality, we tend to employ the predictable minimum of methods of evaluation for convenience' sake. Recently, research and development in this line has made rapid progress, with types and details of methods of evaluation advancing from, for instance, open-loop control to pick up simple functions of the vehicle to closed-loop control to observe the vehicle's reaction to disturbance and further to closed-loop plus feedback control as a system with consideration given to human factors, though it is said that in principle there are many unknown parts essential to the driver.

As a result, the vehicle seems to be assured of safe running with significantly higher accuracy and a wider range. Among these, the performance specifications for the handling and stability required of ESV may be considered one of the most advanced methods of evaluation. Although it is a rather high-degree method, its prime objective is to evaluate the handling and stability essential to the vehicle, and it seems not to be entirely satisfactory as a method of evaluation of the accident avoidance ability, evaluation with stress laid on more practical operations of the driver based on specific analysis of accidents. If the problem is viewed only in the light of car performance, it is impossible for a vehicle to compensate for the consequences of careless or reckless operations by the driver, as shown by actual examples of accidents, no matter how excellent its performance might be. Rather, in some cases, the effect of improved performance may act adversely in the overall safety system, including even the driver's psychology.

To solve this problem, it is necessary to establish, in parallel with the establishment of the handling and stability essential to the vehicle, the accident-avoidance characteristics based upon harder facts, with human operations as the main constituents as well methods of evaluation thereof; it is believed that we should complete each specific task performance, especially using data from analysis of accidents.

From such a point of view, we have been subjecting the Honda ESV to task performance tests aiming chiefly at studying accident avoidance ability, of which I would like to introduce today those conducted on tires and brakes.

THE JAPANESE TECHNICAL PRESENTATION

SECTION TWO PART 1

TIRES AND BRAKES

AKIHIRO IRIMAJIRI, *Staff Engineer*
Honda Research and Development Company, Ltd.

I would like to describe and report, though partially, on the tires and brakes. First, let me discuss the vehicle handling evaluation of the run-flat tire from a comparative and analytical viewpoint.

The purpose of equipping vehicles with run-flat tires may be generally divided into the following: to reduce the incidence of tire failures and to reduce the incidence of accidents as the result of tire deflation; by enabling the vehicle to proceed safely to the nearest service shop without changing tires.

It is our opinion that when one of its tires deflates, the vehicle should be able to meet the following requirements:

1. Drivers should be able to drive safely.
2. The vehicle should be capable of running a significant distance at a speed of 50 km/h, the lowest speed allowable on Japan's highways, without replacing the deflated tire with a spare.
3. The vehicle should be able to maintain sufficient handling to ensure a minimum of safety during such running.

Items (1) and (2) above may be solved as problems of the tire itself. However, as the third one, the characteristics of vehicles are closely associated with those of tires. From such a viewpoint, we have been conducting tests with actual vehicles.

For this test, the vehicle was equipped with run-flat tires of support type. It is a vehicle with about 1,500 pounds weight and 2,200 mm wheelbase, the same specifications as those for the Honda ESV.

The test method is shown in Figure 1. With one of its front or rear tires deflated, the vehicle turns so that the deflated wheel serves as an outer wheel, and during such turning the vehicle's static margin is obtained from the steady-state yaw response.

To obtain the static margin from the steady-state yaw response the equation, as shown in Figure 2, is employed, in which case the dimension of the static margin comes to the reciprocal of the square of the velocity.

Figure 3 shows the results of tests conducted at 40 km/h and at 80 km/h, both with a lateral acceleration of 0.4 g. Compared with the case where the tire inflation pressure is normal, the static margin changes noticeably when one of the tires deflates. Also it changes observably as the speed on the axis of abscissa changes. Tests conducted for reference purpose with the vehicle equipped with non-run-flat tires show a similar trend when one of its tires is lowered in the inflation pressure; however, it could be said that the result shows better performance than in the case where the bursting occurs on one of its run-flat tires.

As the velocity increases, the static margin will decrease if one of the front tires is deflated, and it will increase if one of the rear tires is deflated, thus

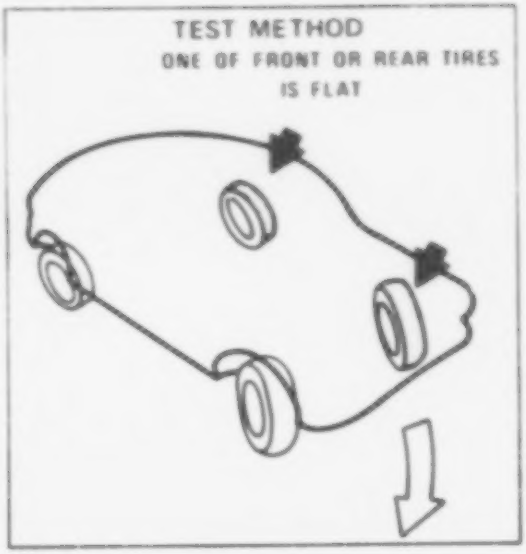

Figure 1

$$\text{STATIC MARGIN} = \frac{V - \frac{r}{\delta}\cdot \ell}{\frac{r}{\delta}\cdot \ell \cdot V^2}$$

V : FORWARD VELOCITY

$\frac{r}{\delta}$: YAW RATE GAIN

ℓ : WHEEL BASE

Figure 2

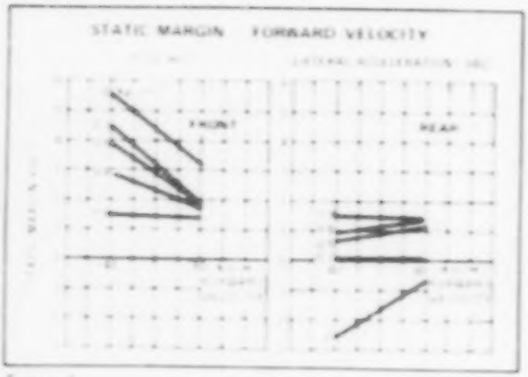

Figure 3

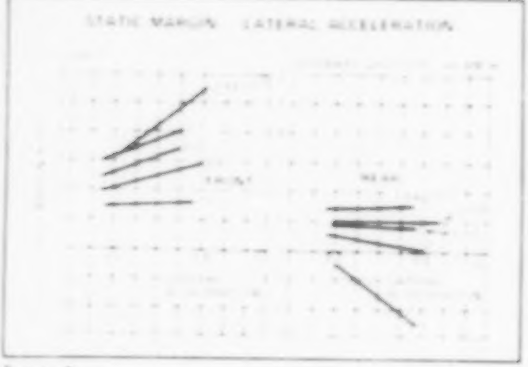

Figure 4

approaching the characteristics of when the inflation pressure is normal. As the vehicle speed increases, the tire characteristics will approach those with the normal inflation pressure, whether the tire is completely deflated or is only lower in the inflation pressure.

Figure 4 shows the results of tests conducted at 40 km/h, with a lateral acceleration of either 0.2 g or 0.4 g. With the change in the lateral acceleration, the static margin will vary little when the inflation pressure is normal, while it will increase if one of the front tires is deflated or lower in the inflation pressure and it will decrease if one of the rear tires is in such a condition.

When one of the rear tires is deflated, and the lateral acceleration increases, the static margin will increase substantially in the negative value, thus placing the vehicle on the danger side.

It can be seen from this fact that cornering power will reduce the run-flat tire at a relatively low slip angle.

Figure 5 shows that the characteristics of the vehicle with run-flat tires tested this time differ significantly from those of vehicles with normally inflated tires. As the graph shows, the cornering-force characteristics of the run-flat tire tested in a laboratory is the same as those for tires used for the actual vehicle test. The effect of load changes is shown at the right and that of velocity changes at the left. In either example, linearity seems to be lost at low slip angle.

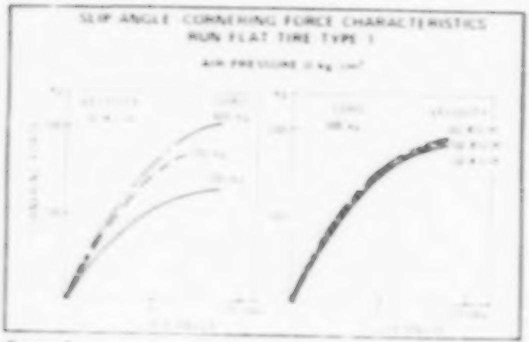

Figure 5

Figure 6 shows the cornering-force characteristics of another type of run-flat tire tested in the laboratory. Apparently, the characteristics shown by the two types of run-flat tires are remarkably different from each other. The vehicle's behavior will also differ when it is fitted with run-flat tires of different types.

The characteristics of run-flat tires mentioned previously are not the only factor affecting the vehicle's behavior. The characteristics of the vehicle as shown in Figure 7 will also be another important factor.

The amount of tire deflection resulting from tire bursting will change according to the load on each wheel. By acting in conjunction with the vehicle's load and roll stiffness distribution between front and rear. And, the deflated tire will also change in the characteristics as a result of the driving/braking force applied.

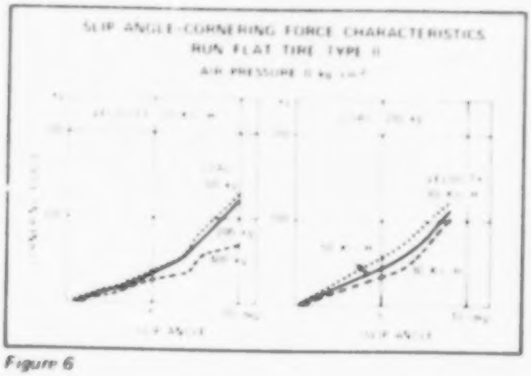

Figure 6

VEHICLE CHARACTERISTICS

1. LOAD DISTRIBUTION BETWEEN FRONT AND REAR.
2. ROLL STIFFNESS DISTRIBUTION BETWEEN FRONT AND REAR.
3. LOCATION OF DRIVING WHEEL.

Figure 7

The adoption of new tires of this type will require not only improvement of the tire itself, but also establishment of a safety system for the vehicle involving the tire as a factor, or an overall safety system including instructions for the man driving the vehicle.

As for the tire, it must be improved so that when it is deflated its characteristics approach those of tires inflated to the specified pressure. However, since any deflated tire cannot retain the same deflection and characteristics that it has when its inflation pressure is normal, the vehicle's characteristics should be improved so as to compensate for the adverse effect resulting from such difference.

Moreover, even with such improvements, the run-flat tire cannot be a panacea. Drivers have to be instructed of the tire's limited performance; otherwise, the run-flat tire would constitute a safety hazard.

I would like to describe concretely the antiskid **brake** system to be installed on the Honda ESV. Brakes are one of the important items in the vehicle's accident avoidance ability and a variety of measures is employed to upgrade their performance level. Among such measures, the antiskid brake system is one of the major items. Specifically speaking, the Honda ESV is equipped with an antiskid brake system acting on four wheels, so as to assure (1) directional stability, (2) handling, and (3) reduced stopping distance.

The main features of this system are:

1. A combined type utilizing both wheel acceleration-deceleration control and slip ratio control.
2. Being compact in size, it can be used on small cars, since the system requires no extra power source for its hydraulic control.
3. Friction coefficient of the road surface is also taken into consideration as a factor in the judgment on the wheel control.
4. A higher directional stability is assured with the adoption of independent front and low-select rear control.

As for vehicle performance, the system has a marked effect on handling and directional stability; therefore, I am going to describe its effect on the stopping distance (Figure 8).

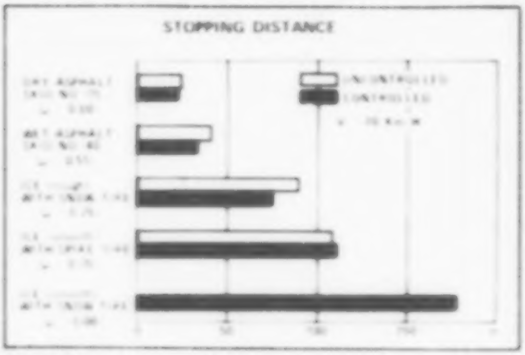

Figure 8

The stopping distance was measured on the following five different surfaces.

1. On dry asphalt with a skid number of 75 (μ=0.80)
2. On wet asphalt with a skid number of 40 (μ=0.55)
3. On rough ice using snow tires (μ=0.25)
4. On smooth ice using spike tires (μ=0.15)
5. On smooth ice using snow tires (μ=0.08)

(NOTE: The friction coefficients referred to are those obtained at the time of wheel locking with the surface and are given only for reference purposes.)

As shown in Figure 9, the stopping distances of the vehicle with the skid control system are approximately 10% shorter on dry asphalt and 15% on wet asphalt pavement and on rough ice than those of the vehicle without the system. (However, on smooth ice, such a comparison cannot be made because of a remarkable difference in the directional stability.)

Figure 10 shows the stopping distance vs. initial velocity curves. Within the low speed range the difference in performance between the vehicle with the system and the vehicle without is not necessarily distinguishable, but it is distinct in the high speed

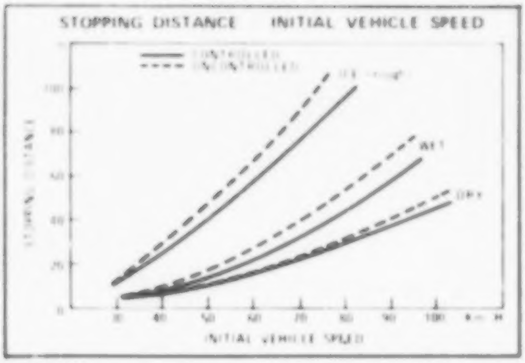

Figure 9

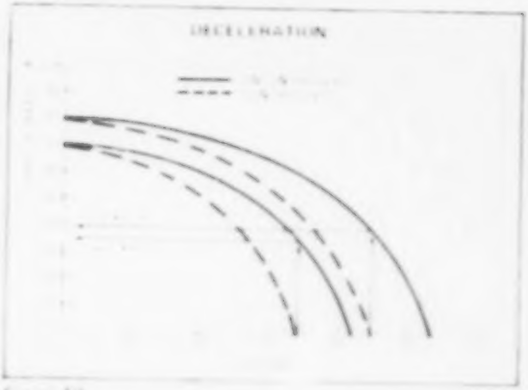

Figure 11

road. When the velocity is high or the friction coefficient is low, the stopping distance will be so long that the difference in distance on the braking becomes a very important problem.

When viewed from another angle, the difference in the stopping distance that I mentioned will give you an understanding of how great its effect is; that is, when one vehicle comes to a stop, the other one with a longer stopping distance is still running, and your viewing angle is to find out at what velocity it is running at that time. Examples given in Figure 11 are those showing the results of tests conducted on wet concrete pavement. According to the graph, the difference is so remarkable that when the vehicle with the skid control system comes to a complete stop after application of the brakes at initial velocity of 70 km/h and 80 km/h is formed and the vehicle without the system is running at 34.5 km/h and at 39.0 km/h respectively.

We believe that such a four-wheel anti-skid brake system is a system which is very effective in other respects. It is our intention to proceed with further studies.

THE FEDERAL REPUBLIC OF GERMANY
TECHNICAL PRESENTATION

SECTION TWO

PART 2

INTRODUCTION

HELMUT WAGNER, *Deputy Assistant Secretary*
Federal Ministry of Transport

I am very pleased to be able to introduce the papers by the German delegation. As you probably know from previous conferences, the initiative for the work on the ESV is up to the German industry supported by the Association of the Insurers and Underwriters of Industry. Today we will present two papers which are centered around the main emphasis of our work during the past years.

THE FEDERAL REPUBLIC OF GERMANY TECHNICAL PRESENTATION

SECTION **TWO** PART **2**

PROGRESS OF THE ESV DEVELOPMENT AT DAIMLER–BENZ

DR. HANS SCHERENBERG, *Member of the Board of Management*
Corporate Chief Engineer
Daimler-Benz AG

In order to place some perspective on developments of the Daimler-Benz ESV program since the Washington conference, I want to summarize what we feel are the most important results of the first three International ESV Conferences.

ESV PROGRAM BEFORE THE THIRD INTERNATIONAL ESV CONFERENCE

The first conference, in January 1971, in Paris, mainly provided basic comments of the governments and automobile manufacturers participating in the ESV Project. Statistical or technical information was not correspondingly voluminous and was not related to an actual ESV. A certain doubt was evident as to the targets and feasibility of the quantum jump in the development of automobile technology sought by the DOT.

The second conference, in October 1971, in Sindelfingen, also dealt with program statements of the governments. The different interpretations of the objectives of the international ESV program by the various governments were already becoming visible. But above all, the technical presentations of the participating automobile manufacturers brought to light extensive new information and considerable differences in respect to the goals and targets of ESV development. The Sindelfingen Conference became a forum for very important discussions, the contents and results of which were centered around two basic, but diverging, concepts.

- The philosophy of the American Government, which establishes an overriding priority for passive safety and crashworthiness, is to make great technical progress in a single, large step.
- The philosophy of the European automobile manufacturers is to follow a step-by-step plan which we feel is more realistic and efficient; we do, however, put equal emphasis on accident avoidance and crashworthiness. Daimler-Benz summarized this philosophy at the Sindelfingen Conference when we proposed a five-point reorientation of the ESV program, as follows:

1. Discussion of the targets which can be reached in the near future with justifiable utilization of personnel and financial capacity.
2. Establishment of a step-by-step plan, also for the experimental safety vehicle project, which sets reasonable objectives within sufficient periods of time.
3. Promotion and extension of biomechanical research.
4. Compilation of statistical data which, in particular, will clarify the relationship between accident severity and impact speed.
5. Establish a well-founded cost/benefit analysis.

Following this, the DOT announced a reorientation of the ESV program.

At the third conference in May 1972, in Washington, no such reorientation was evident, although the ESVs presented up to that time confirmed, without exemption, all of our doubts concerning weight and cost. Thus, the need for reorientation became even more clear and we have extensively explained the reasons. Once again DOT promised a reorientation. Yet it was not expected that this could be accomplished in a short time.

TARGETS OF DAIMLER-BENZ ESV DEVELOPMENT BETWEEN THE THIRD AND FOURTH CONFERENCE

Lacking any formal changes in the program, we decided to establish our own project concept. It was based on the two following criteria:

1. The level of passive safety in today's mass production vehicles is represented by the 30-mph collision into a rigid barrier. This safety level can be economically achieved in high quantity production with reasonably small tolerances. The fact that the number of traffic victims still remains too high is due less to technical deficiencies than to the fact that vehicle occupants do not use available and approved occupant protection elements, such

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as safety belts, or that rulemaking authorities do not enforce compulsary use of belts.

2. ESV development during 1971 and 1972 has shown that improvement of passive safety up to a barrier impact speed of 50 mph is indeed physically possible, but can only be achieved through a huge technical effort, which is easily recognized in terms of the additional vehicle weight — for the Mercedes-Benz ESF 13, for example, 700 kg or a 50% increase over the original weight. In this case, the upper limit of technical feasibility is obviously approached without prior proof of need.

But the upper limit of economical feasibility is clearly exceeded. Such a vehicle can never be an economical solution for the majority of people throughout the world. Furthermore, it would cause secondary economic burdens, such as additional fuel consumption, higher emissions and need for additional traffic and parking areas.

The logical consequence was a research program which investigated physical and economical vehicle design possibilities throughout the large range between the technology of today and the original aims of the ESV program. We decided to initiate this project by using the new Mercedes-Benz S-class, which contains all new mass-producible elements of occupant safety and represents the most recent stage of the continuous Daimler-Benz safety improvement effort.

The result of this is the new Mercedes-Benz safety research vehicle ESF 22 (see page). Within the scope of our research activities on ESVs we created our own step-by-step plan. This was intended to investigate, in detail, the unknown territory between the advanced technology, as it is applied in our production today, and the goals of the original U.S. specifications.

The ESF 22 is a result of this work and is not intended to meet all of the original specifications. Based on the assumption that the specified maximum deceleration values permitted for the test dummies actually represent the tolerable maximum for the human body, the ESF 22, with its new easy-to-handle and comfortable safety belt system, offers chances for survival in frontal collisions against a rigid barrier up to 40 mph.

Nevertheless, the research vehicle shows 660 pounds of additional weight. A further increase by 140 pounds would be inevitable should the drive train and the driving gear have to withstand a durability test. As a result of the requirements for higher engine output with increased weight, both fuel consumption and exhaust gas flow will be increased by about 10 to 15%. For these reasons, the ESF 22 cannot be regarded as an economically feasible solution for mass production.

The ESF 22 is based upon our new models 450 SL and SLC. These models already incorporate, as standard, all-important elements of occupant protection, according to the latest status of the continuous development of safety at Daimler-Benz.

However, the ESV research work at Daimler-Benz with solutions such as the ESF 05, 13 and 22 not only widened our knowledge but also raised a large number of questions. These mainly concern problems of accident occurrence in road traffic, tolerance limits of the human body, important physical relations during the collision, the severity index, aggressiveness, current test procedures and measuring equipment, in particular the test dummies.

Still pending is the question relating to the overall effectiveness of such a program devoted solely to the technical aspects of the automobile, and not directly and distinctly aimed at the other traffic partners — driver and road. Obviously, these cannot be disregarded if a maximum of total safety is to be reached. In its safety research Daimler-Benz will continue to consider solutions to these open questions a major task, and we shall make added efforts towards these goals.

Thus we find ourselves deep in a research program requiring continuous critical re-evaluation of results which had apparently been assured. However, I may emphasize one result in particular — the increase of three-point safety belt effectiveness. The belts in today's mass-production cars are designed for the 30 mph frontal collision. These belts could help prevent more than half of all severe and fatal injuries, if each occupant would wear them. The new experimental belts offer corresponding survivability up to 40 mph frontal collisions. It is justified to assume that they could prevent a considerable additional number of severe and fatal injuries. At the same time, modern retractor belts are easy to handle, convenient to wear, permit sufficient freedom of motion and are reliable.

In view of these factors, it would be inexplicable if the majority of car occupants would refuse to wear belts in the future. Furthermore, such a rejection would not be tolerable, and so, only one consequence remains — a legal requirement to use belts. The legislators of many countries are seriously investigating this question and one can only hope, for the sake not only of the individual car occupant but also the total economy, that it will be decided quickly and positively. The need for passive restraint systems could then probably be reconsidered.

Less positive are the results of our reflections on vehicle aggressiveness. This concept was presented at the first ESV conference with justified fear of the results which may occur during collisions between relatively light cars and the cars of an eventual American ESV generation three times as heavy. Ideas on the determination and rule-enforced limitation of aggressiveness have developed too quickly, for example, Appel's theory, which inevitably would lead to the installation of large hydraulic buffers, or theories advanced in the "Carter paper," which proposes to design all front structures with one and the same crush resistance. The result would be vehicle lengths which would never be accomodated by any rational traffic system.

We consider research on aggressiveness as not having reached a stage where design and engineering solutions can be approached. For the time being, a statistical analysis of road accidents with regard to mass ratios of colliding vehicles is more important.

Only the frequency distribution of accident-vehicle masses can show between which lower and upper value will be the range of vehicle masses in which an aggressiveness index should be meaningfully determined and limits applied. For it is indeed economic Utopia to require complete compatibility among all vehicles, from the 1,000-pound city car up to the 40-ton truck/trailer combination. In addition, the definition and application of an aggressiveness index has to be flexible so that it permits a large variety of design and engineering.

TASKS OF THE FUTURE

In our opinion, this example demonstrates very clearly that the full consequences of the ESV program are not now recognizable. It has led to new ideas, but at the same time has raised many new questions. This is not to say that it has not been worthwhile, but now these questions must be precisely formulated and investigated before we attempt to establish any new program.

These questions concern not only technical principles and engineering solutions but more often the accident occurrence in road traffic. The most important premises of active and passive safety, such as ascertained knowledge about frequency and severity of accidents, are not available. Again and again we must ask the following questions:

- Which are the most frequent accident modes?
- Which physical quantities are indicative of the severity of an accident — the equivalent speed in the collision against a fixed rigid obstacle? the actual velocity change? the deformation of the vehicle?
- Is the frontal collision against the rigid plane barrier indeed the most important accident simulation or, as Cornell notes, are frontal collisions against rigid poles more representative?
- Shall we replace the fixed rigid obstacle by
  a. a fixed barrier having a deformable impact surface?
  b. a movable barrier with a rigid impact surface? or
  c. a movable barrier with a deformable impact surface?
- Shall a yielding barrier be regenerative, i.e. cheap and reproducible, or an actual vehicle?
- In which mass value range should the movable barrier be variable?

Obviously, we recognize more and more that these questions have to be answered first before we can establish a new set of specifications. In our opinion, the tasks for the future will therefore have to begin with the investigation of these partial problems. In particular, we must develop

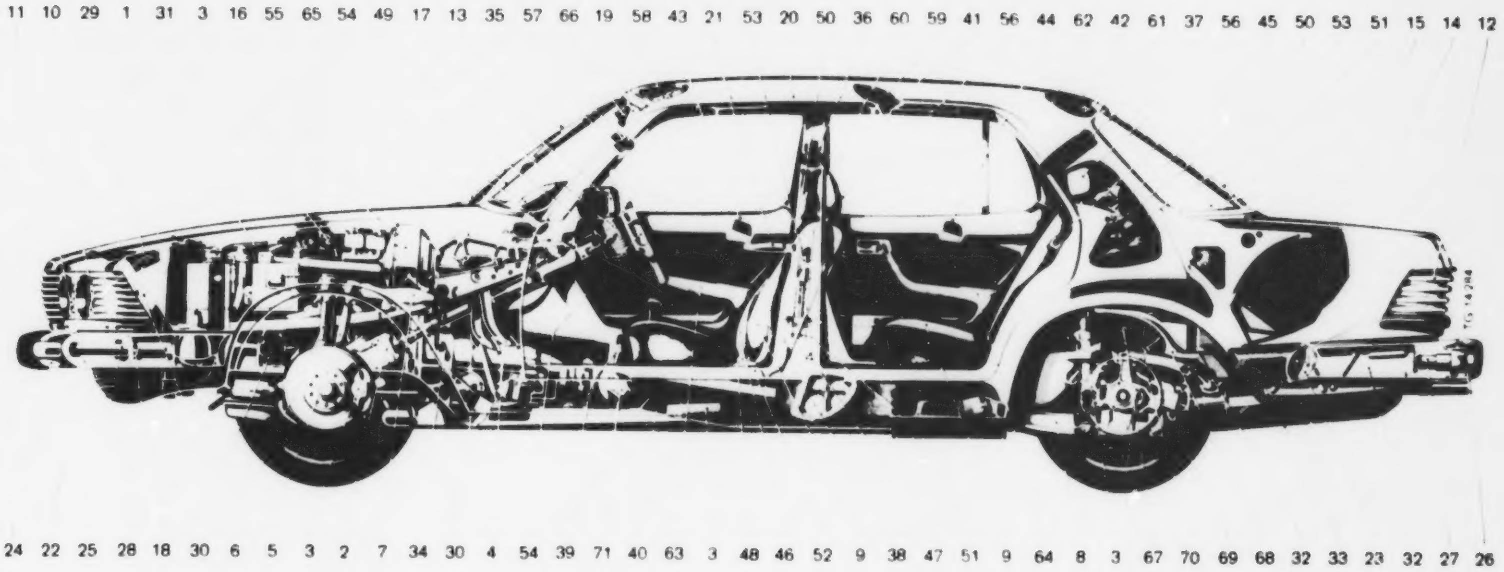
- A new priority order of accident modes
- An accident severity index
- An aggressiveness index
- New representative test procedures
- New injury criteria
- New reliable measuring devices with emphasis on dummies, and as often stated before
- A well-founded cost/benefit analysis

Only after this research has produced results is a synthesis into a new set of specifications possible. This, of course, does not mean that vehicle safety can not be improved in the meantime. New knowledge from basic research will immediately be employed at least for the development of subsystems — in other words, a continuing process of adaption and optimization will occur and, as I already said in the earlier conferences, intermediate results will show up in the cars of current production.

We should, therefore, discontinue the separate development of total ESVs and instead, continuously promote safety by profound and coordinated investigations in all critical areas. If standards are to be introduced at all, the principal aim should be the introduction of a uniform system of standards in all countries. Today, the diverging regulations compel us to develop and build many different modifications of a car type.

Mercedes-Benz
Experimentier-Sicherheits-Personenwagen
ESF 22

Mercedes-Benz
Experimental-Safety-Vehicle
ESF 22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24 22 25 28 18 30 6 5 3 2 7 34 30 4 54 39 71 40 63 3 48 46 52 9 38 47 51 9 64 8 3 67 70 69 68 32 33 23 32 27 26

ESF 22

Technical data

General data

Body style	four door sedan
Number of passengers	4
Overall height, unloaded	1430 mm
Overall width	1870 mm
Overall lenght	5240 mm
Wheelbase	2860 mm
Seat height, unloaded front	970 mm
Seat height at rear	860 mm
Track width front	1525 mm
Track width rear	1505 mm
Turning circle diameter	11.40 m
Boot space	0.579 m³
Vehicle curb weight	app. 2025 kg
Gross vehicle weight rating	app. 2425 kg
Tank capacity	app. 96 l
Tires with tubes	215/70 HR 14

Active Safety

Driving safety

1 Engine

Electronic fuel injection
Emission control acc. 1973 USA specifications

Number of cylinders	V8
Total displacement	4520 cm³

Engine output – SAE net HP/rpm 195/4500
Max torque – SAE mkp/U/min 36.5/3000

2 Steering

MB recirculating ball type power steering

Brakes

3 Service brake Mercedes Benz/Teldix A.B.S.
4 Foot operated parking brake

Suspension

5 Front axle
Dual wishbone with progressive anti dive geometry
6 Hydraulic suspension
7 anti roll bar
with zero offset steering
8 Rear axle
Semi trailing axle with
anti pitch feature

Environmental safety

Seating comfort

9 Seats designed with respect to orthopaedic findings, stiffness and damping tuned to the vibration characteristics of the vehicle, porous upholstery

Air conditioning

The air conditioning preserves the best physical condition of the occupants by
sufficient oxygen supply (ventilation)
heating or cooling
optimum temperature distribution
favourable humidity

Vibration and noise characteristics

Significant reduction of vibration and noise is achieved by proper design of major components and body as well as by suitable insulation and damping

Visibility Safety

Headlights

10 4 headlight system with halogen bulbs
11 Headlight wiper and washer

Rear lights

12 Anti fouling control by ribbed surface
Warning signal when vehicle is stopped
Marker lamps in all doors when opened
13 Wiper and Washer for windshield
with large wiped area, no lifting at high speed,
two wiping frequencies and interval control
14 Rear window heated
15 Vertical wiper blade traversing horizontally for rear window
16 Device for self cleaning of side windows

Safety color

Increased recognition through color combinations

Operational safety

By locating and designing the controls, anthropometric and ergonomic points of view have been considered
all parts used have been taken from standard cars
17 Outside rearview mirror, adjustable from the interior

Passive Safety

Exterior safety

(Safety for pedestrian and cyclists)
Compact exterior styling and design
18 Entire frontal area of foam material
17 Folding outside mirror
19 Integral radio antenna
20 Rain channel made of rubber
21 Rounded door handles
22, 23 Bumper side parts of foam material and mounted on the body without gap

Interior safety

Exterior deformations

Front bumper system
24 Bumper mid portion
25 Regenerating shock absorbers with 175 mm stroke impact speed up to 8 km/h without damage to the vehicle
22 Bumper side extensions
Rear bumper system
26 Bumper mid portion
27 Regenerating shock absorbers with 75 mm stroke impact speed up to 8 km/h without damage to the vehicle
23 Bumper side extensions

Front structure

In case of head on collision with speeds up to 65 km/h the kinetic energy of the vehicle is absorbed by properly defined deformation of the
28 Lower front crossmember
29 Upper front crossmember
30 Lower front sills
31 Upper front sills
25 velocity sensitive absorbers

Rear structure

In case of rear end collision with speeds up to 50 km/h the kinetic energy of the vehicle is absorbed by properly defined deformation of the
32 Rear sills
33 Spare wheel and spare wheel trough
27 velocity sensitive shock absorbers

Passenger compartment

The passenger compartment consists of a rigid structure surrounding the occupants, while protecting them against the exterior forces thus enabling the other components of interior safety to function properly (exterior and interior deformation, restraint system)
34 Fire wall crossmember
35 Windshield pillar
36 B pillar
37 Rear pillar
38 Side sills
39 Drive train tunnel
40 Cross member below front seat
41 Side roof rail
42 Rear header
43 Door reinforcements front door
44 Door reinforcements rear door
45 Door locks
21 Door handles with counterbalance weights

Restraint systems

Passive restraint systems for high impact speeds
Driver
46 Three point safety belt system with
47 automatic retractor and
48 three webbing load limiting devices
49 Airbag system
54 in the energy absorbing steering wheel hub
Front seat passenger and rear seat passengers
50 Three point safety belt system with
51 automatic retractor and
52 three webbing load limiting devices and
53 emergency preloading device
Additional restraint system
54 Steering wheel with energy absorbing device (see also Interior Deformation)
55 Safety dash board (see also Interior Deformation)
9 Front safety seats with side mounted seat adjustment tracks, both having locking mechanisms
56 Adjustable head restraints

Interior deformation

Interior deformations are those which occur at the inner surface of the passenger compartment while absorbing the kinetic energy of occupants and thus reducing the impact forces, provided that the restraint system does not suffice to prevent an impact against the inner compartement surfaces
55 Safety dash board (see also restraint systems)
54 Safety steering wheel (see also restraint systems)
57 Padding on "A" Pillars
58 Padding on front header
59 Padding on side roof rail
60 Padding on "B" Pillars also containing energy absorbing tube elements
61 Padding on "C" Pillars also containing energy absorbing tube elements
62 Padding on rear header
63 Front door panels with mounted armrests of foamed material
64 Rear door panels with mounted armrests of foamed material
65 Front windshield laminated safety glass, bonded
14 Rear window laminated safety glass
66 Impact detaching inside mirror

Fire prevention

Preventive measures

67 Locating of the fuel tank above the rear axle with great distance to the exhaust system and
68 bulk head between both systems
Fuel tank and fuel hoses are so designed and arranged that mechanical stress during crashes is largely avoided

Emergency measures

69 Another device at the fuel tank prevents fuel leakage in case of an abnormal vehicle position and reduces any increase in gas pressure
70 A valve system prevents the tank contents from leaking in case of damaged fuel hoses

Passenger compartment

The materials used have a burning rate less than 100 mm/min
71 Fire extinguisher is mounted below the front edge of the driver's seat ready for instant use

This manifold development variety, and the necessity for reliable compliance in equally segmented mass production, unnecessarily ties up the time, energy and talents of thousands of qualified engineers and technicians. If we could have uniform regulations, a large portion of this capacity would be available to carry out effective research and development which is not purely oriented to state programs, but includes our own concepts of beneficial trends and goals for progress in automobile technology. Industry research was by far not the last to provide essential foundations for legislation and could continue to do so in the future.

Finally, I would like to deal with a problem which not only is related to the ESV development but which, today, is a goal destined for rulemaking (i.e. the definition and evaluation of vehicle handling properties).

Good handling properties are not verified by a remote-controlled vehicle, which performs perfectly in extreme maneuvers which a driver could never carry out. Even less extreme maneuvers, which are performable by drivers, are only valuable for research purposes if conducted mechanically by remote control. They can never be used for judging actual handling qualities.

Tests at the handling limits constitute only one portion of the correct handling features. Indeed, the majority of tests during the development of an automobile are performed in the medium range, because most drivers do not voluntarily carry a vehicle to the extreme limits of its handling capabilities. Of course, they can get carried there if the vehicle does not have good handling properties in its medium range. The behavior in the car's boundary range thus is expected to function only as a rescue possibility.

Basically, besides the vehicle's yaw response to steering movements the quality of the information channels also is important for the driver. Handling safety largely depends directly upon how quickly and exactly the driver feels and sees each shade of vehicle motion to which he must adapt the next steering action and correction movement. The English word "handling" designates exactly this complex interaction between driver and car.

A skilled test driver can distinguish good and bad handling properties and, in some cases, indicate the underlying reasons. But today, we can not yet describe, mathematically, how an automobile should respond and how the nature of the information channels must be to become optimal for the majority of drivers. It is even less possible to analytically set the limits which should not be surpassed, so that the handling quality can be designated as "still safe."

Many of our own analytical investigations have shown that the objective of measurement and evaluation of handling with any kind of recording apparatus is almost unachievable. The reason for this is the overwhelming number of combinations of modes and variations of driving manuevers and of environmental and operational conditions to be included.

Through this fairly recent work, it has been proven again what an outstanding analyst and evaluator the human being is. He must perform a huge quantity of comprehensive individual tests during one single test drive under mixed conditions; he must evaluate them; and he must store the results and then extract the significant features to be improved. We do not know how many weeks or months of analytical work would equal this one ride.

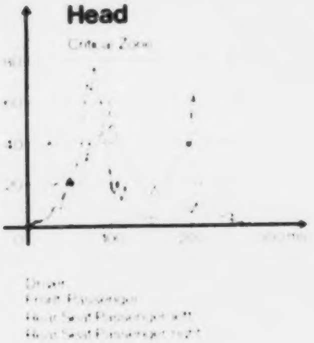
In addition, we do not know whether analytical methods can comprehend handling to the extent that they can be a suitable tool for a law enforcing, inspecting, or compliance-testing agency. During the recent past, our expectations in this respect have been reduced, rather than increased. We don't state this on the basis of a preconceived tendency against any rulemaking or standardization; but we are unhappy about this. Because analytical methods, as a means for developing handling features, would relieve us first from the difficulties of the continuous need to compare vehicle-to-vehicle, and second, from the variations of subjective evaluations. We would willingly accept some additional expense if there were an analytical method. But, if oversimplified tests with wide validity for good handling characteristics are to be conceived, then we honestly cannot see the faintest image of land even with the strongest telescope.

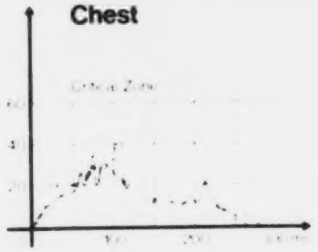
We, therefore, urge DOT and the participating governments to consider the situation in the field of car handling very profoundly and sincerely and to start work in this area with the greatest scepticism. We should mutually agree that, today, the faith in the judgment on handling properties by a test driver is the best we have and that any hope for another method of evaluation will not be forthcoming for a long time to come. We would very much regret it if rules were to be approved based on insufficient procedures, such as the requirements on handling and stability in the new U.S. specifications for the intermediate ESV.

RESULTS OF THE FINAL RELEASE TESTS OF THE ESF 22 CONDUCTED AT THE CLOSE OF DEVELOPMENT

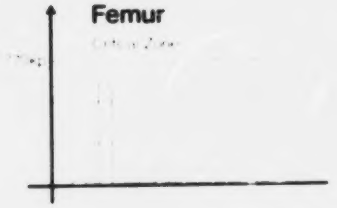
Deceleration values or forces are plotted on the vertical axis; time is plotted after the commencement of the collision along the horizontal. The border line values of 80 g for the head and 60 g for the chest and the pelvis, according to today's status of accident research, may be exceeded for a total of 0.003 seconds, without having to expect serious injuries. For the load on the femur, 1,700 pounds equals 770 kg are permitted. Beyond these values, in the critical range, heavy injuries are possible, depending on the constitution of the occupants and the duration of deceleration.

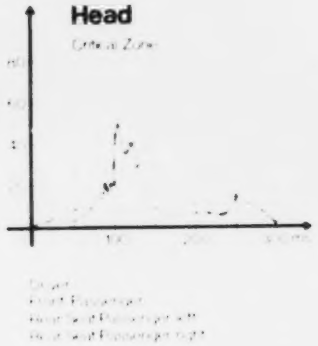
All Daimler-Benz test results are mostly well below these allowable limits, with the exception of the value for the driver's chest during the side collision vehicle-to-vehicle. The excess of 10% possibly can be traced back to the actual collision speed, which was 10% higher than required (33 instead of 30 km/h). Because of measuring instrument failure during the angled, frontal-barrier collision test, both the values of the driver's chest and the co-driver's head could not be recorded.

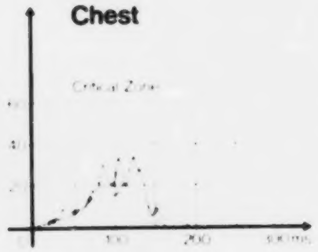




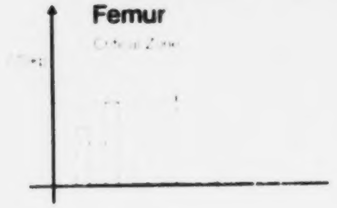
Frontal Collision at 65 km/h against Fixed Barrier

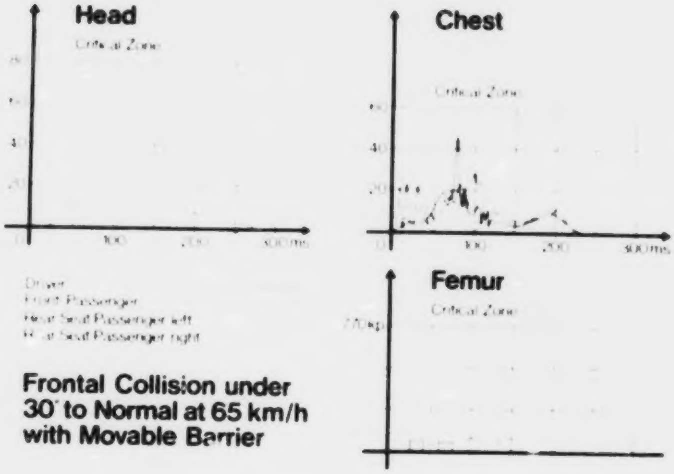






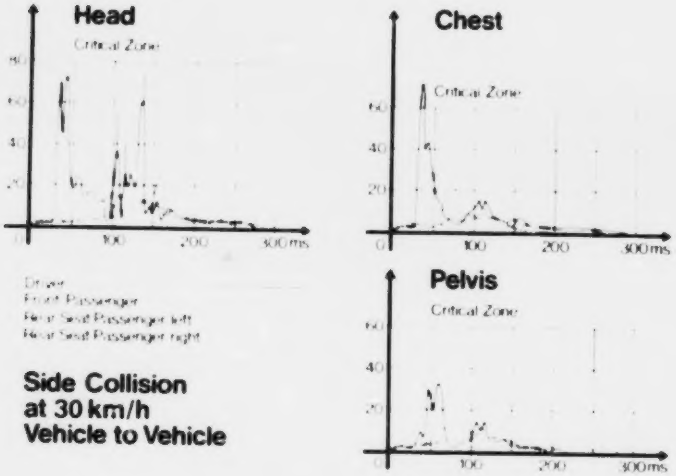
Frontal Collision at 50 km/h against Fixed Pole





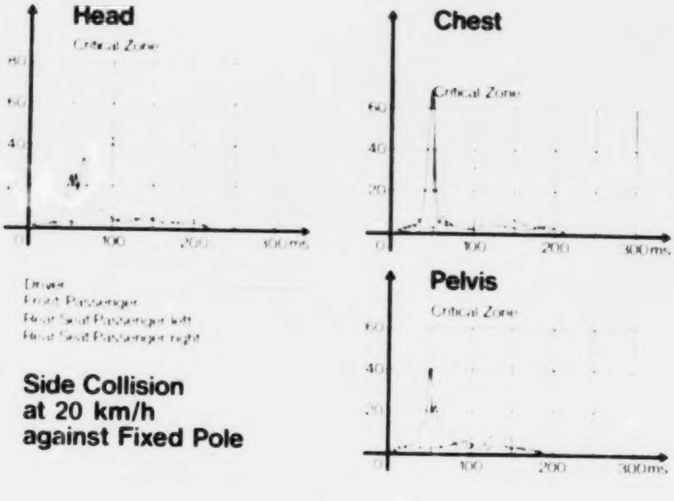
Driver
Front Passenger
Rear Seat Passenger left
Rear Seat Passenger right

Frontal Collision under 30° to Normal at 65 km/h with Movable Barrier



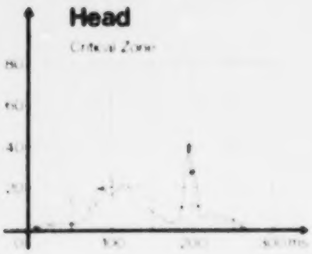
Driver
Front Passenger
Rear Seat Passenger left
Rear Seat Passenger right

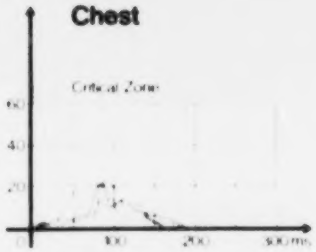
Side Collision at 30 km/h Vehicle to Vehicle

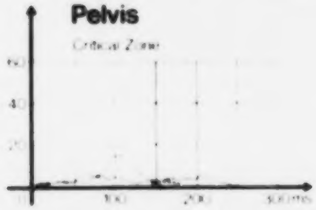


Driver
Front Passenger
Rear Seat Passenger left
Rear Seat Passenger right

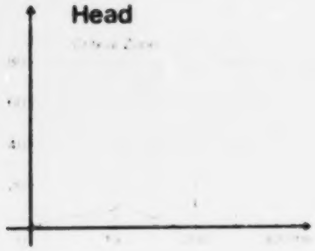
Side Collision at 20 km/h against Fixed Pole

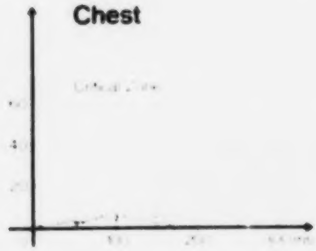


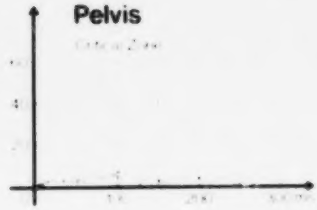




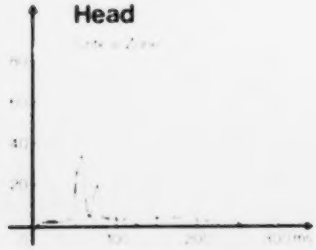
Rearend Collision
at 50 km/h
with Movable Barrier

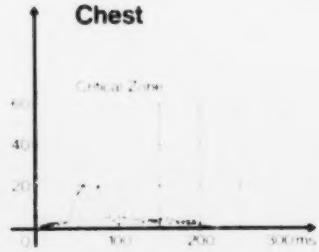


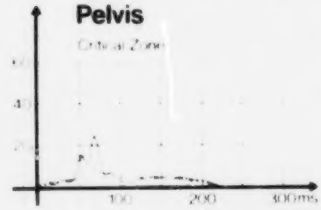




Rearend Collision
at 20 km/h
against Fixed Pole







Drop Test
from 0,5 m

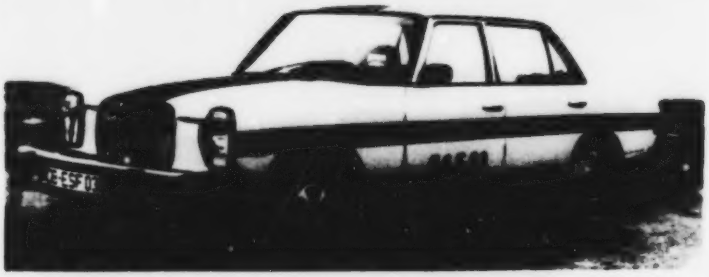
Subsystem Testing and Evaluation

ESF 01

ESF 01a

ESF 02

ESF 03
Sindelfingen
26.05.71



ESF 04

ESF 04a

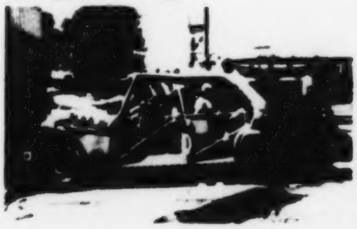
ESF 06

ESF 07

ESF 08

ESF 09

ESF 10



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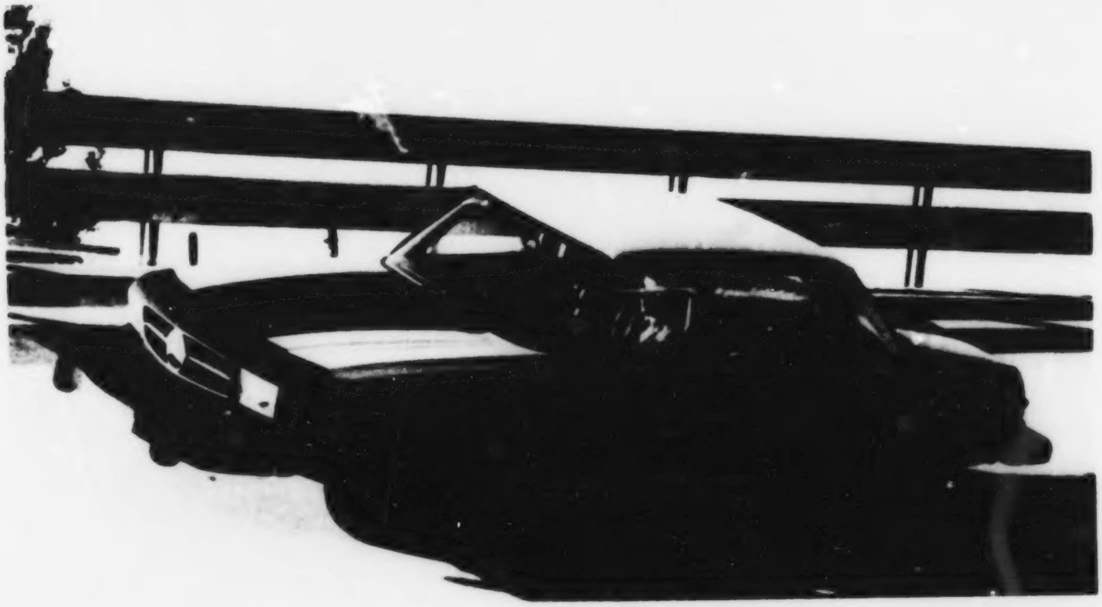

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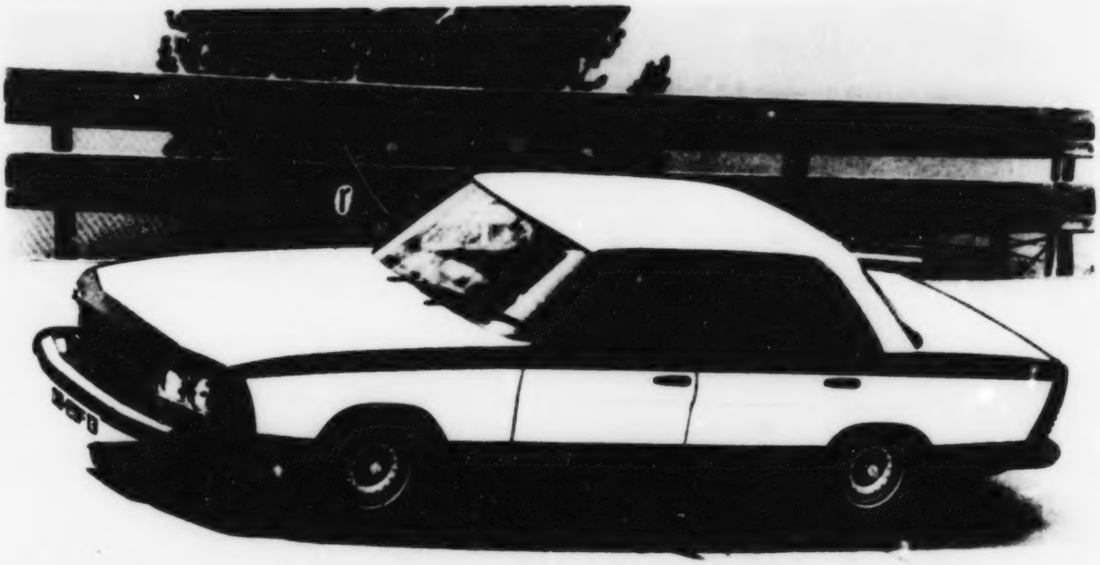

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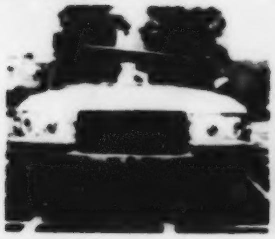

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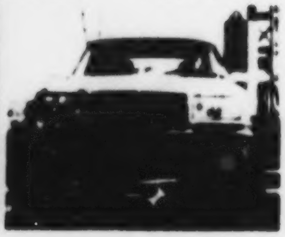

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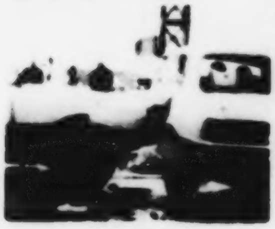

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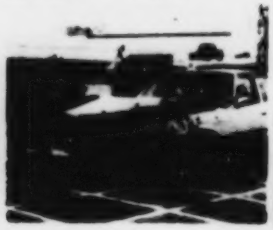

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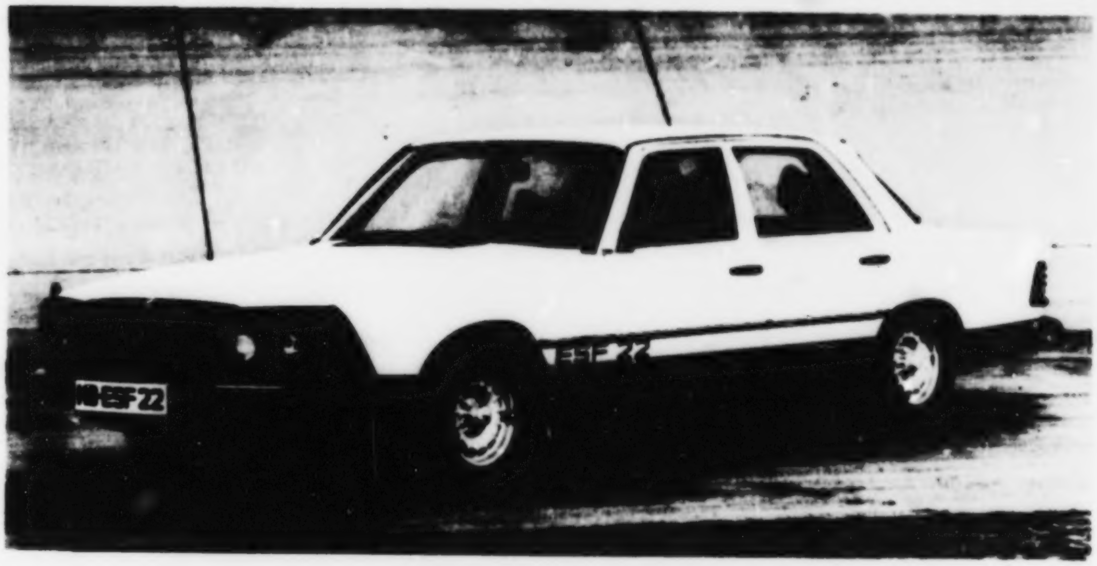


THE FEDERAL REPUBLIC OF GERMANY TECHNICAL PRESENTATION

SECTION TWO

PART 2

PROGRESS REPORT ON OPEL'S SAFETY VEHICLE ACTIVITIES

BERTHOLD FELZER *Project Manager*
Adam Opel AG

FIRST PHASE

After publication of the German Technical Specifications for a 2,000 pound ESV in late 1970, Opel began its activities in this field in March 1971. We quickly found that we were on the path of creating a monster. We had designed a car with the interior space of a small-size vehicle but with the exterior dimensions and weight of a car weighing almost 3,000 pounds (Figure 1). This was a paper and component study and we refer to it as our Phase One ESV activities.

SECOND PHASE

In July 1971, we started our second phase, with the study of actual traffic accident statistical data. Material from different sources, as well as from our own investigations, were used to develop a tool for adjusting the specifications. Based on the results of this study, we revised the ESV requirements and set new priorities (Figure 2). We also learned that approximately 70 percent of all passenger cars in Europe belong to the lower weight car class up to 2,000 pounds of curb weight.

Besides basing our ESV work on actual statistical, traffic accident data and the small car class, we also directed our design objectives to the present high-volume production technology and the utilization of conventional materials. Our weight studies were particularly critical, because increased weight increases cost, lowers performance, increases fuel consumption and increases the braking requirements.

Our progress prior to the Second International ESV Conference in Stuttgart-Sindelfingen included several test cars which had been built and evaluated. We contributed to this conference by reporting on the status and progress of our activities.

Our main efforts were concentrated on developing light-weight, low-cost, high energy-absorbing structural elements with the characteristics necessary to accomplish our performance objectives (Figure 3). A potential solution was found with polyurethane hard foam-filled structural elements (Figure 4). Opel reported on this work during the Third International ESV Conference in Washington.

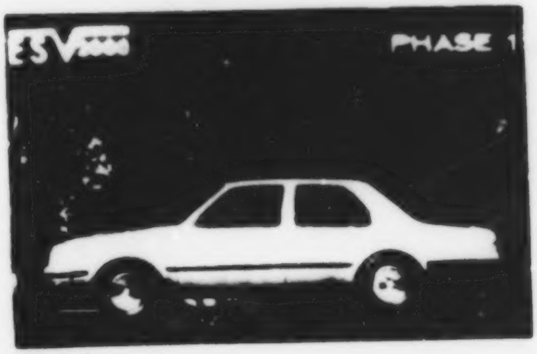


Figure 1

ESV 2,000 – PRIORITIES:

1. FRONTAL IMPACTS
2. ROLL OVER
3. SIDE IMPACTS
4. REAR IMPACTS

Figure 2

GENERAL

Following the conference, further traffic accident data obtained from different countries, from the German Car Insurance Association (HUK) and from results of our benefit/cost investigations indicated to

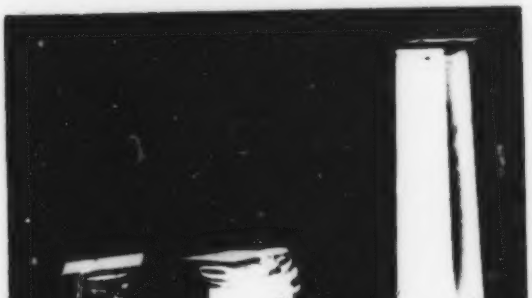


Figure 3

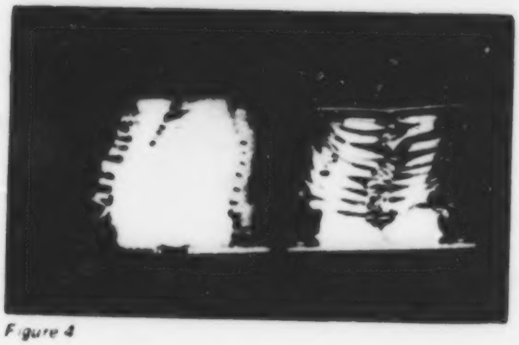


Figure 4

us that we should readjust our objectives and reduce the barrier test requirement from 50 mph to 40 mph.

This was also based on NHTSA data showing that the present 30 mph barrier test would give survivability in 38 percent of the fatal accidents, 40 mph would give 74 percent survival and 50 mph 93 percent survival (Figure 5). This showed us that by going from 30 mph to 40 mph the survivability could almost be doubled, whereas the survivability by going from 40 mph to 50 mph would only be increased by another 19 percent. A similar study is required with regard to the expected reduction of injuries.

The weight and energy absorption requirements would increase in some proportion to the following: Increasing the barrier test speed of vehicles from the presently practiced 30 mph to 40 mph under curb weight condition increases the energy ratio to 178 percent (Figure 6). Taking that into consideration, to crash test the vehicle fully loaded at 40 mph increases the energy ratio to about 250 percent.

ENERGY RATIOS

- 30 mph BARRIER C.W. 1.0
- 40 mph BARRIER C.W. 1.78
- 40 mph BARRIER F.L. ~ 2.5
- 50 mph BARRIER F.L. ~ 4.0

Figure 6

The structure of the same fully-loaded vehicle in a 50 mph test would have to manage the energy close to four times the energy of the present 30 mph test. This indicated to us that we would reach our proposed objective faster by concentrating on the very significant step of 40 mph fully loaded first, and reviewing the ultimate value of even higher barrier capabilities later.

From the different traffic accident data available after the last ESV conference, we learned that more than 92 percent of all frontal accidents, with injured occupants, occur in an impact speed range up to 60 km/h, which equals approximately 37 mph (Figure 7).

THIRD PHASE – OCCUPANT PROTECTION STATUS

We use the term Opel Safety Vehicle (OSV) in this third phase to denote that we intend to have an independent, flexible program.

After readjusting our design objectives after the Washington conference, our OSV program was grouped into three major segments:

- component development program
- basic test program
- prototype program

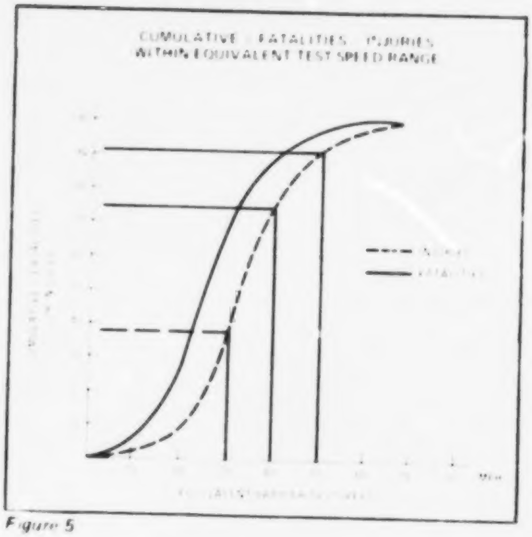


Figure 5

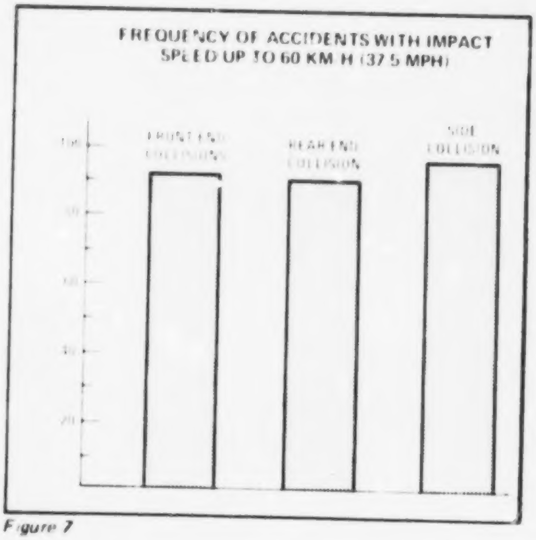


Figure 7

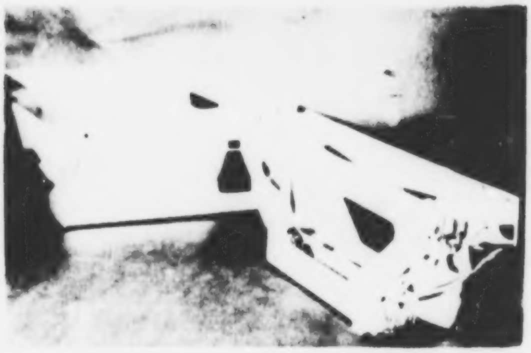


Figure 8

Component Development Program

The component development program is aimed at feasible solutions in each safety-related area, which could be incorporated in safety vehicle prototypes after extensive testing with pretest vehicles. This program has no final target date in the near future to enable a continuous optimization process in the various fields. The main efforts are being concentrated on:

- vehicle structures
- seat development
- steering spindles
- bumper development
- rear view systems
- lighting systems

Concerning the work on improved vehicle structures, we are in a phase of trying to convert the simple foam-filled elements we introduced to you during the last ESV conference into vehicular dimensions. Besides the continuation of the test work with single foam-filled structural elements, we also made some progress in the testing of simulated front end structures (Figure 8).

Special attention is also being given to seat development (Figure 9). As all of you in this business know, rear and side impacts cause some problems with respect to seat shape, seat structure (including seat back), seat adjustment mechanism and fixtures to the car (Figure 10). We are evaluating different seat designs in our basic test program to study their behavior during real car impacts. The development of different seat belt systems, as well as head rest designs, goes along with the seat development program (Figure 11).

Several proposals to improve the steering spindle design are under investigation (Figure 12).

The peripheral bumper approach (Figure 13), as well as other bumper concepts, are being evaluated with respect to crash behavior, pedestrian friendliness and production feasibility.

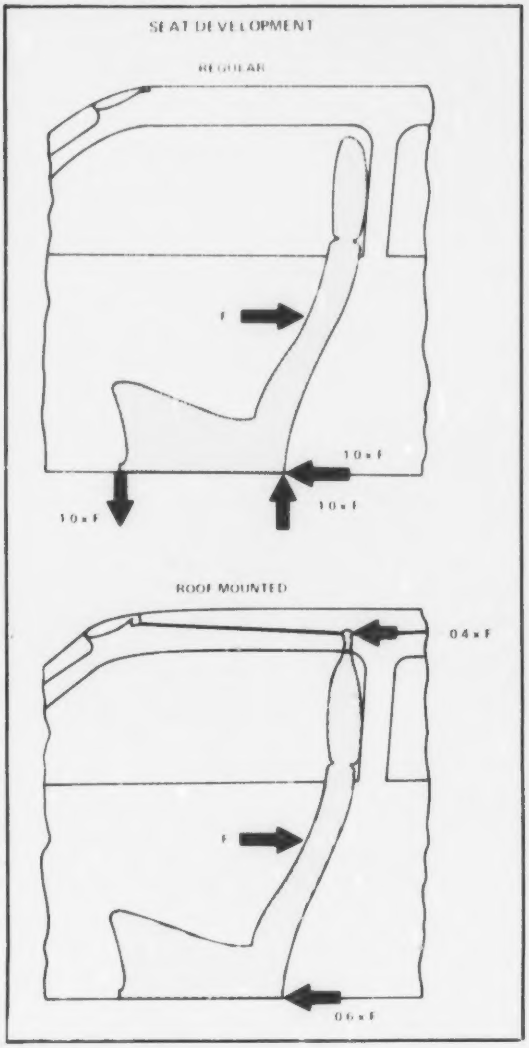


Figure 9

Rear view systems to improve the rearward view, as well as reduce the so called "dead angle," are being tested (Figure 14).

Variable intensity lighting systems and different arrangements of tail, brake and hazard-warning lights are being evaluated (Figure 15). A light check-out system for the driver also is being included in our investigation.

When we have satisfied ourselves that we have developed components which will meet our safety, weight and cost objectives, we intend to incorporate these components into our OSV prototype vehicles.

Figure 10

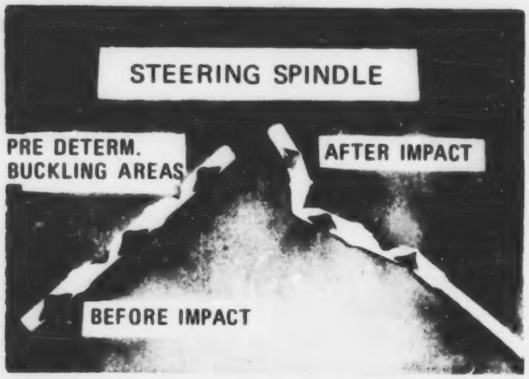


Figure 12

Figure 13

Basic Test Program

The purpose of our basic test program was mainly to evaluate and illustrate the impact of our inhouse OSV specifications on regular production vehicles (Figure 16). From the results of these tests we were able to direct our development work to specific areas (Figure 17).

Besides these tests, we also did some simulation work involving actual traffic accidents, including multiple crashes, to study the occupant dynamics during the different crash phases. The influence of the different load conditions on crash behavior has been tested in 40 mph barrier impacts using production vehicles (Figure 18).

It is interesting to note that by increasing the energy content to 178 percent, compared to the presently-practiced 30 mph barrier impact, no noticable damage to the passenger compartment was

Figure 11

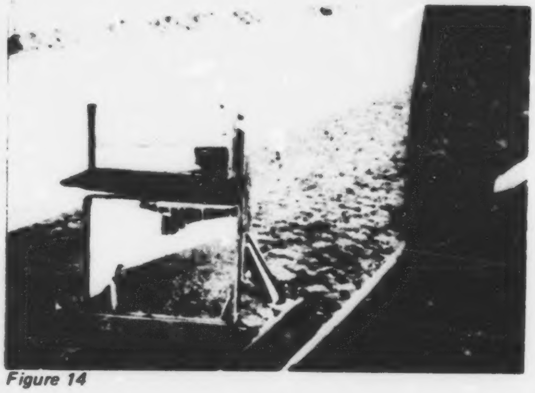


Figure 14

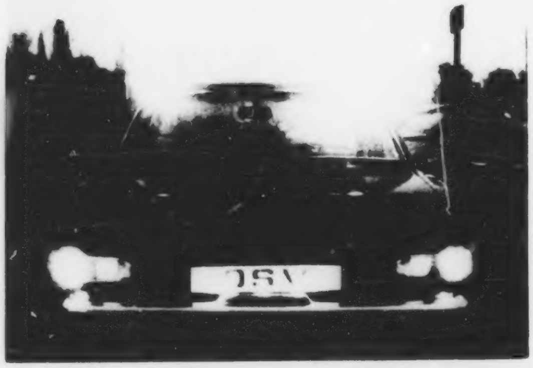


Figure 15

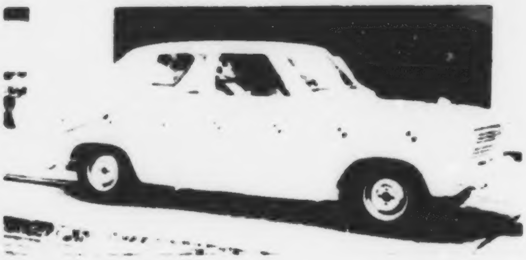


Figure 16

registered (Figure 19). The same vehicle, fully loaded, representing an energy content increase to about 250 percent, showed a different picture (Figure 20).

This made clear the challenges we have to face, because 40 mph barrier capability with fully-loaded vehicles is one of our targets.

OSV Prototype Program

The layout and design work for two prototype vehicles is almost finished. The two vehicles will be used only for testing purposes in the occupant-protection field. Acceptable solutions from the com-

Figure 17

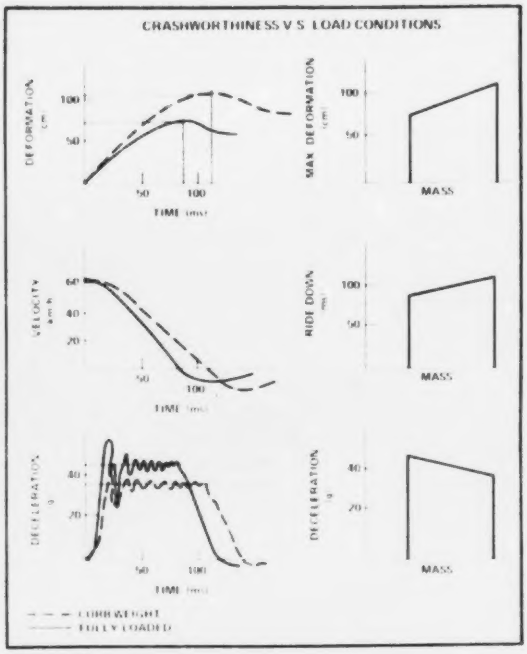


Figure 18

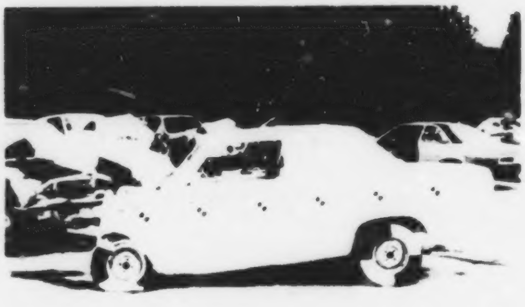


Figure 19

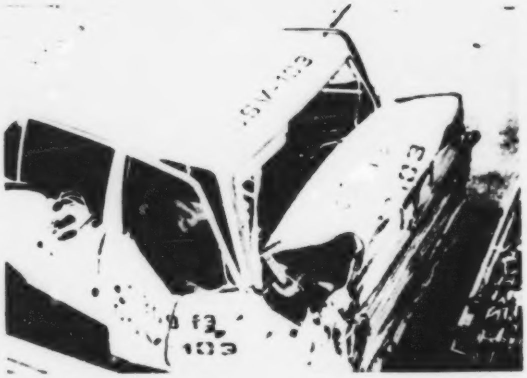


Figure 20

OSV – TEST PROGRAM

- 40 mph BARRIER FRONT
- 30 mph REAR MOVING BARRIER
- 30 mph CAR TO CAR SIDE IMPACT
- 20 mph SIDE POLE (MOVING)
- 30 mph FIXED POLE FRONT
- 2 ROLL OVERS

Figure 21

ESV BRAKING REQUIREMENTS

• SERVICE BRAKE (40% RATED LOAD)	
PEDAL FORCE NOT TO EXCEED	40 kp
PEDAL FORCE MEASURED	25 kp
• PARKING BRAKE (FULLY LOADED)	
30% GRADE FORWARD NOT TO EXCEED	40 kp
MEASURED	33 kp
30% GRADE REVERSE NOT TO EXCEED	40 kp
MEASURED	27 kp

Figure 22

ponent development program are going to be integrated into these vehicles and tested with the cars in the various impacts planned (Figure 21). Additional prototype vehicles are scheduled to be built after the test program has been completed, to make use of the test data for further improvements.

Accident Avoidance

At the last conference we showed some braking as well as riding and handling tests. In the meantime, we have finished the evaluation work and are in a position to state that all braking requirements, as well as most riding and handling specifications, can be met (Figure 22).

The lower weight cars in our production line showed a constant understeer tendency and a steering returnability well within the given limits (Figure 23). The lateral acceleration of 0.65 g could be met with regular tires and even exceeded by using radial tires. Steering control sensitivity created no problems and the overturning immunity tested on a pylon course can also be met (Figure 24). The J-turn specifications with a steering wheel deflection rate of not less than 500 degrees per second and a deflection of not less than 180 degrees has been found unrealistic for most of the European vehicles without power assistance and lower overall steering ratio. We propose to use a steering wheel deflection rate of 360 degrees per second and a total deflection of 120 degrees (Figure 25).

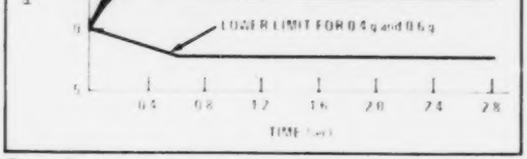


Figure 23

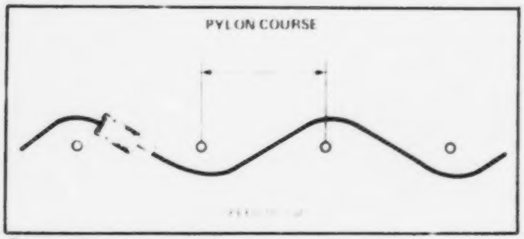


Figure 24

J–TURN REQUIREMENT PROPOSED:

- STEERING WHEEL DEFLECTION RATE 360 deg/sec
- TOTAL DEFLECTION 120 deg.

Figure 25

Benefit-Cost Study

As mentioned earlier, we also performed benefit-cost studies for the purpose of defining the most effective and economical way to protect the occupants. Unrealistic safety measures cannot be of benefit to the driving population, because vehicles extensively increased in price would not be attractive to the majority of the customers and hence would not help in reducing the death and injury rate on the roads.

After we had determined the different benefit-cost ratios for the various safety measures, we compared these ratios, within the same applicable fields, to select the most efficient and promising way to solve the problem.

SUMMARY

In summary, we have used the term OSV to denote that we are on an independent program and intend to stay flexible with our inhouse specifications for a safety vehicle. This will enable us to come up with meaningful, optimized solutions.

Figure 26

Our objective priorities of the different steps are being determined by evaluation of various traffic accident statistical data available, whereas the different measures towards meeting our objectives are being based on continuing benefit-cost studies. It should be mentioned that the benefit-cost calculations are based on the German domestic situation and are not necessarily applicable to other countries.

The work with our foam-filled structural elements is progressing and we are trying, now, to convert these simple elements into vehicle-structure dimensions (Figure 26). We realize that this is a time-consuming process, due to the unknown influence of the various parameters, such as the right combinations of foam, sheet metal and profile dimensions. We know that this process will take quite a bit of concentrated effort to develop applicable solutions for high volume production processes. Besides defining the right combinations for high energy absorbing vehicle structures, the behavior during an angular impact as well as the influence of the engine in the crash picture must also be studied carefully.

SECTION TWO PART 2

EXAMPLES OF THE DEVELOPMENT OF ESV SUBSYSTEMS – PORSCHE KG

DR. HANS-HERMANN BRAESS, *Chief*
Basic Research and Exploratory Development
H.C.F. Porsche KG

In order to improve traffic safety, tested safety elements should be incorporated in production cars as early as possible. This requirement is, in fact, not contradictory to the development of complete experimental safety vehicles which are intended to conceive and test the novel car concepts of the future.

Like some other automobile manufacturers, Porsche started to continuously improve the safety of its cars quite a number of years ago. For instance, Porsche cars reached values of 0.7 g for lateral acceleration as early as 1954, while even today's ESV specifications ask only for 0.65 g. The twice interrupted and rejoined steering column is another safety element which Porsche has been using for quite a number of years.

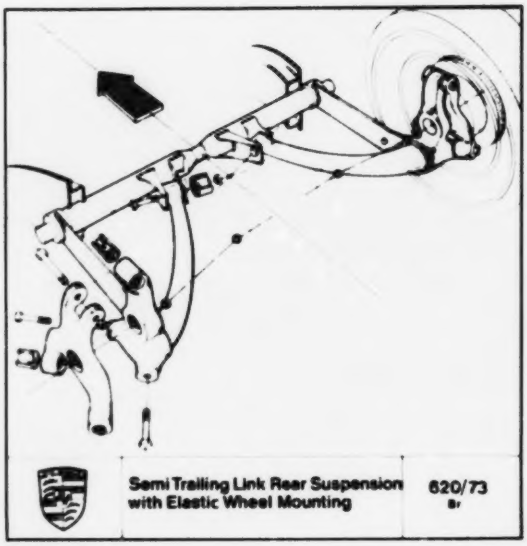
The first example of automotive research which is going to be dealt with today refers to active safety. The pertinent specifications contain specific requirements that modifications of lateral acceleration, engine load conditions and payload must not induce any significant changes in the understeer-oversteer behavior.

These are some of the reasons which led us to investigate, among other things, the automatic adjustment of the wheel positions to the different driving conditions by introducing certain elasticities in the right places without affecting other characteristics of the wheel suspension in question, such as the position of the roll center, kinematic modifications of toe-in, etc.

We were envisaging a solution in which the wheel was to be mounted to its suspension, not rigidly, but via specially positioned elasticities.

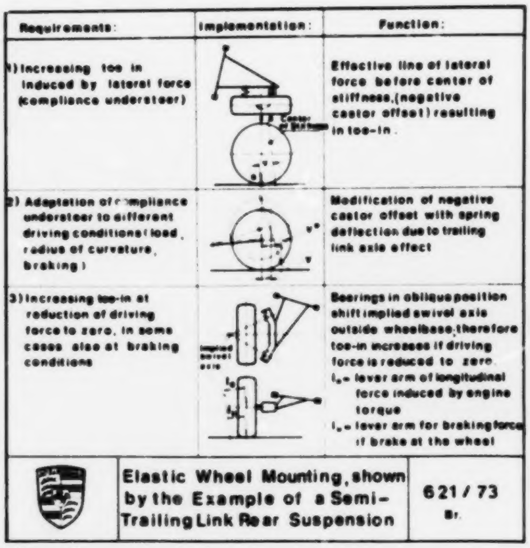
For a better understanding of the implementation shown in Figure 1 we are going to start with the investigation of the ways to fulfill the individual requirements.

The upper part of Figure 2 shows, in the example of a semi-trailing link rear suspension, a way to obtain a generally desirable compliance understeer by locating the center of stiffness behind the rear axle line.



Semi Trailing Link Rear Suspension with Elastic Wheel Mounting 620/73 Br

Figure 1



Requirements:	Implementation:	Function:
1) Increasing toe in induced by lateral force (compliance understeer)	Center of stiffness	Effective line of lateral force before center of stiffness, (negative castor offset) resulting in toe-in.
2) Adaptation of compliance understeer to different driving conditions (load, radius of curvature, braking)		Modification of negative castor offset with spring deflection due to trailing link axle effect
3) Increasing toe-in at reduction of driving force to zero, in some cases also at braking conditions	implied swivel axis	Bearings in oblique position shift implied swivel axis outside wheelbase, therefore toe-in increases if driving force is reduced to zero. l_a = lever arm of longitudinal force induced by engine torque l_b = lever arm for braking force if brake at the wheel

Elastic Wheel Mounting, shown by the Example of a Semi-Trailing Link Rear Suspension 621/73 Br.

Figure 2

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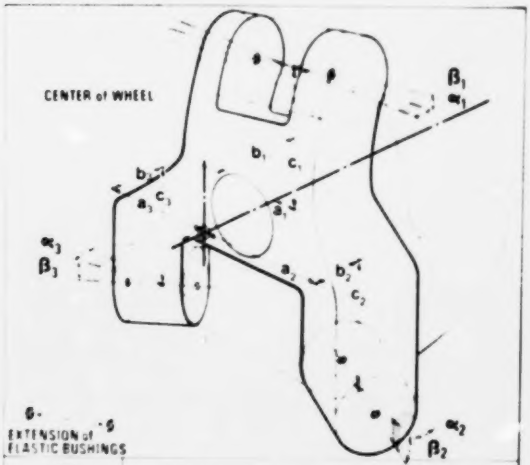
The central part of the figure shows how the value of this compliance understeer can be adapted to the driving conditions by means of the trailing-link axle effect of the wheel in motion. The lower part of the figure demonstrates an increase of toe-in at deceleration by means of a fictive-wheel swivel axis outside the wheelbase. This effect is intended to reduce the frequently observed oversteer in case of engine load changes in curves.

Figure 3 is another view of the bearing configuration shown in Figure 1. The bearings are conventional rubber bearings which are preferably mounted in a forward direction for three reasons: firstly, due to the greater axial elasticity the degree of deformation required to obtain the toe-in modification asked for at changing engine load conditions (requirement No. 3 in Figure 2) can be achieved; secondly, this will help to compensate the rolling roughness of radial tires, i.e. of the high-frequency vibrations coming into effect in longitudinal direction; thirdly, the greater radial hardness of the bearings will allow for an accurate camber angle adjustment of the wheel. In this way the required compliance understeer is achieved in that, contrary to the lower front bearing 3, the bearings 1 and 2 are relatively rigid (rubber hardness approximately 70 Shore compared to 50 Shore). The setting angles α_2 and α_3 will fulfill requirement 3.

Theoretical considerations aiming at optimum bearing configurations especially indicate that the distance b_1, b_2 and b_3 should be small, and the distance a_2 and a_3 should be great.

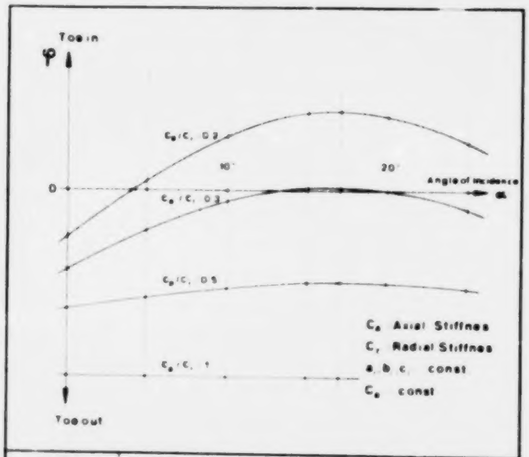


Configuration of the Elastic Bushings 622/73 Br

Figure 3

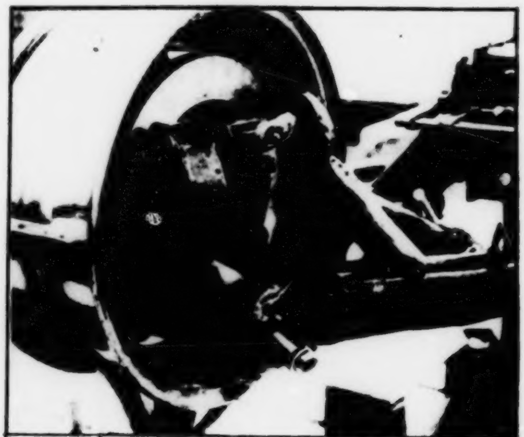
Figure 4 shows a special calculating result, indicating that there are optimum setting angles α_2 and α_3 if a maximum increase of toe-in at deceleration is to be achieved. For the rubber bearings themselves, we have to aim at a low-axial and a high-radial rigidity, especially if the distance b_2 and b_3 cannot be kept small enough for design reasons.

Also for design reasons, the first experimental prototype of the semi-trailing link rear suspension was not executed with two lower and one upper bearings as per Figure 3 but in the embodiment shown in Figure 5.



Analytical Optimization of the Bushing Incidence Angles α_2 and α_3 623/73 Br

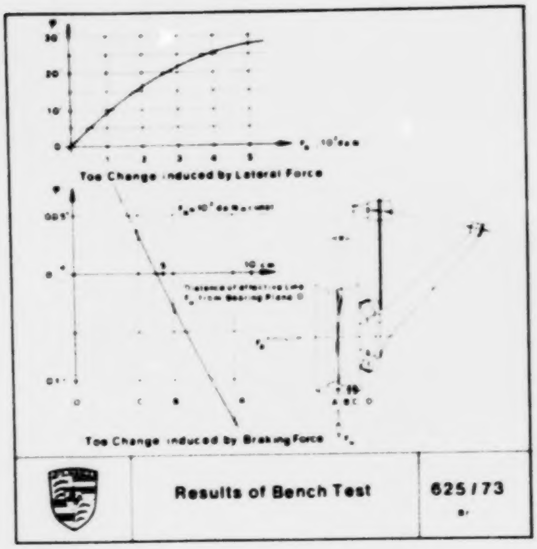
Figure 4



First experimental embodiment of a semitrailing link rear suspension with elastic wheel mounting 624/73 Br

Figure 5

This suspension was first subject to static stress bench tests. While compliance understeer is largely adjustable by a corresponding selection of the hardness of the individual rubber bearings — of which Figure 6 (upper part) is a practical example — as a result of the too large distance of the wheel center from the bearing plane in this configuration (see lower part of Figure 6) the toe-in modification induced by longitudinal force is not yet satisfactory.



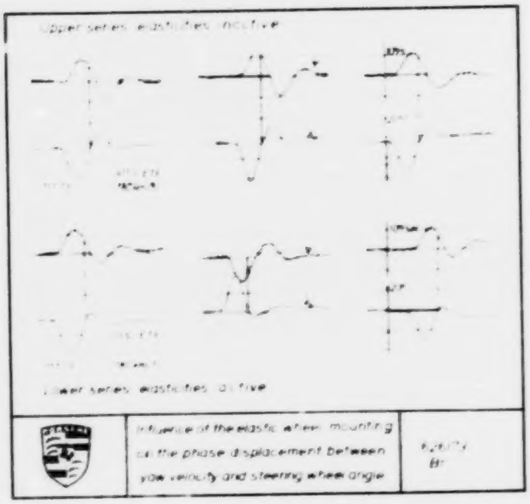
Results of Bench Test 625/73

Figure 6

Road tests on a Porsche 911S showed that compliance understeer will exert a favorable influence both on the stationary and non-stationary understeer-oversteer behavior. The reduction of pressing-on at high speed driving, i.e. the phase lag of the yawing motion compared to the steering wheel movement, is shown as an example in Figure 7.

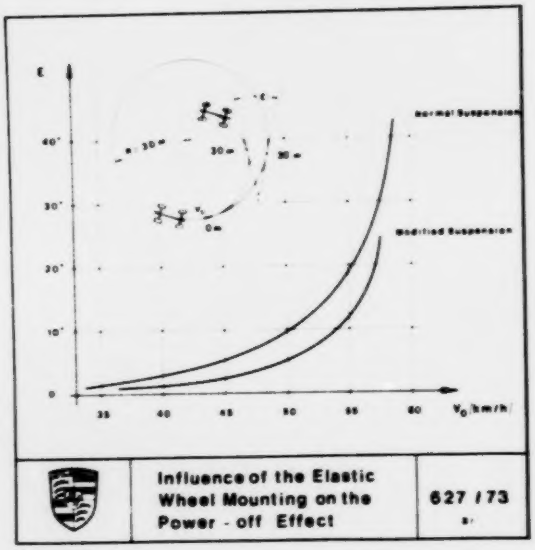
It also will be seen in Figure 8 that the oversteer effect at deceleration could be substantially reduced with the first suspension prototype.

The proposed solution implies another effect: understeering behavior in fast bends of large radii and increasingly neutral behavior as you get into narrower bends. This is induced by the modified trim of the car due to the changing curve radii; in narrow bends the longitudinal component of the centrifugal force will increase the load on the front axle and decrease that on the rear axle. This decrease will reduce the negative castor offset v shown in Figure 2 and thus the value of the toe-in modification at a given lateral force in case of a trailing-link axle effect at rear wheel lifting.



Influence of the elastic wheel mounting on the phase displacement between yaw velocity and steering wheel angle

Figure 7



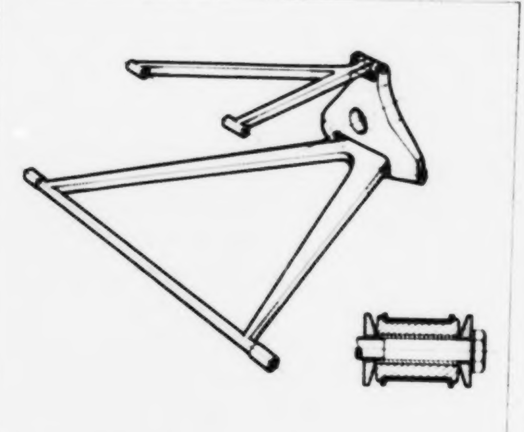
Influence of the Elastic Wheel Mounting on the Power - off Effect 627/73

Figure 8

Figure 9 shows that the proposed configuration of an elastic wheel mounting has numerous applications. This is a double wishbone suspension which will realize the indicated characteristics practically without additional hardware and construction efforts.

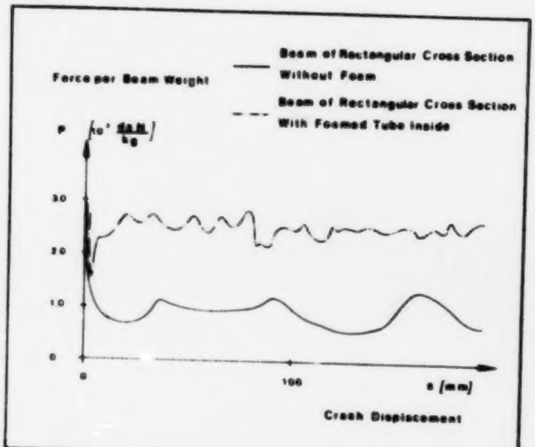
As a number of accidents cannot be avoided, even at a high standard of active safety, the passive safety component should not be neglected. We are now going to deal with an example in this field — the control of crash behavior by means of deformation bodies.

It is known that the energy absorption of sheet metal supports in the body structure is significantly



Double Wishbone Suspension with Elastic Wheel Mounting 628/73 Br

Figure 9



Energy Management of Experimental Crash Beams 630/73 Br

Figure 11

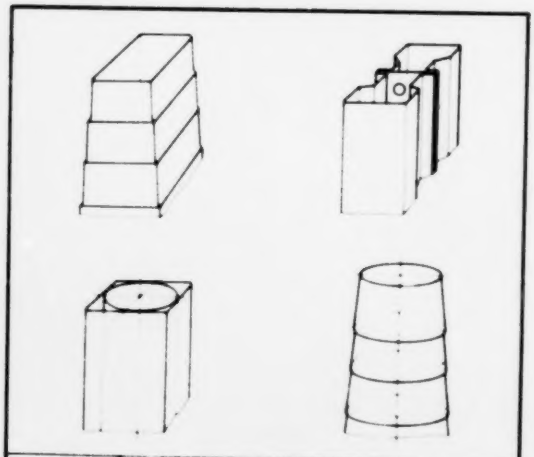
reduced during plastic deformation, including buckling and bending processes. If these secondary phenomena can be kept under control, a high-energy absorption can be obtained even at a small body weight.

Therefore, we carried out a number of static and dynamic deformation tests using different support structures. Some of them — a part of which are shown in Figure 10 — were equipped with internal tubing which, in turn, was filled with P-U-foam.

As shown in Figure 11, supports of this configuration will reach a high-energy absorption at a relatively small weight. This is due to an improved buckling and bending behavior, but also to the fact that the internal tubing will largely increase the number of buckling folds; thus, the energy absorption potential as shown in Figure 12. It was found that the optimum foam density is between 160 and 180 g/1.

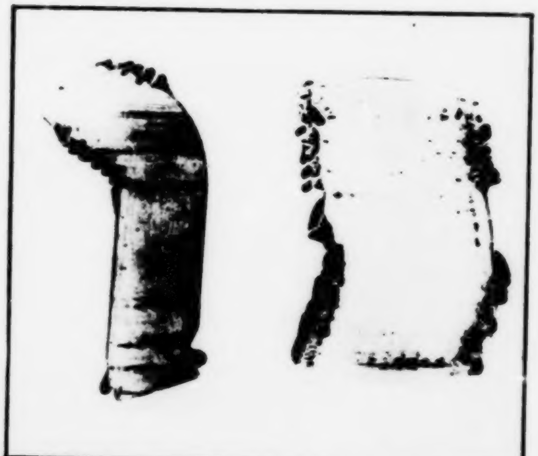
We have demonstrated on film that supports of this configuration will show a controllable deformation and a uniform energy absorption even at oblique impacts (in this case the angle of attack is 15°).

In conclusion, it should be specially noted that the proposed support modifications can generally be



Examples of Experimental Crash Beams 629/73 Br

Figure 10



Comparison of the sections of two deformed crash beams 631173 Br

Figure 12

achieved without major tool modifications, even for present production cars.

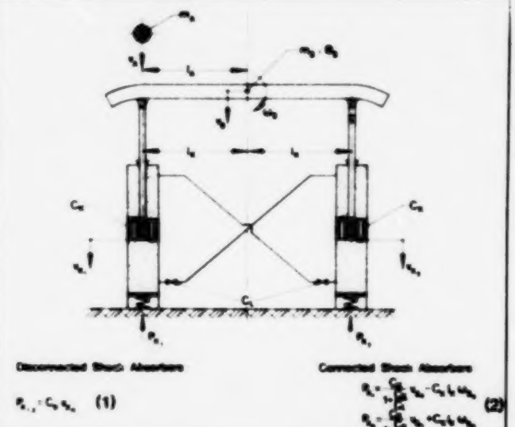
The last example from the development of ESV subsystems deals with hydraulic, shock-absorption systems which will help to obtain an optimum energy absorption at higher impact speeds and to avoid vehicle damage at light collisions.

Statistical evaluations show that oblique impacts from a left hand forward direction happen more frequently than clear, head-on collisions. Because in such cases the attacked body side has to absorb higher forces and deformation energies (and this is especially true for small impact angles), it would seem advisable to select a configuration of the hydraulic shock absorber system which will realize the most uniform stress conditions.

This, in fact, is the intention of the diagonal hydraulic connection between the two shock absorbers which can be seen in Figure 13. Of course, it would seem sensible to study the behavior of the proposed system in theoretical calculation before the hardware realization of the experimental prototypes is started.

The influence of the diagonal connections is neglected for the time being, and the damping properties of the pistons are selected as known, in a way that the damping force is proportionate to the initial piston speed and remains constant for the entire piston displacement; this configuration will attain the highest energy compensation at a given piston displacement (See equation 1 in Figure 13).

In this way, by the superposition of the piston forces coming into effect, due to the piston movements in the same and in the opposite direction, we find the piston force values by equation 2, Figure 13, if the friction influence is neglected. Perhaps we will arrive at a better understanding of the combined effect in the system if the damping forces in the pistons and in the connections are seen as two shock absorbers in a tandem configuration.



Schematic View of a Hydraulic Shock Absorber System with Diagonal Connections

632/73
Br

Figure 13

Figure 14 shows the data based on the schematic view of Figure 13. This is a simplified model, neglecting the movement of the attacked vehicle after the impact to facilitate computations.

Equations 3 and 4 in Figure 14 lead to the initial speed of the shock absorber resulting from the collision theory; it should be noted that these values are independent of the damping properties of the shock absorbers.

In the beginning, the damping value C_k of the pistons (see equations 1 and 2 in Figure 13) is calculated for a centered impact on the disconnected system inducing a decleration of 6 g (corresponding to a total force of 6.12 t).

We arrive at: C_k = 1400 kps/m.

$m_s = 20$ kg	weight of the bumper
$\Theta_s = 0.5$ mkps²	moment of inertia of the bumper
$l_s = 0.5$ m	distance of shock absorbers from car centerline
$m_a = 1000$ kg	impact weight
$V_a = 8$ km/h	impact speed

initial speed of shock absorbers after the impact:

$$v_s(t=0) = v_{s0} = \frac{(1+\varepsilon)\, v_a}{\frac{m_s}{m_a} + 1 + l_a^2\,\frac{m_s}{\Theta_s}} \qquad (3)\ \text{translation}$$

$$\omega_s(t=0) = \omega_{s0} = \frac{-(1+\varepsilon)\frac{l_a m_s}{\Theta_s} v_a}{\frac{m_s}{m_a} + 1 + l_a^2\,\frac{m_s}{\Theta_s}} \qquad (4)\ \text{rotation}$$

in this $\varepsilon = 0$: fully plastical shock
$\varepsilon = 1$: fully elastical shock

CALCULATING THE DAMPING FORCES

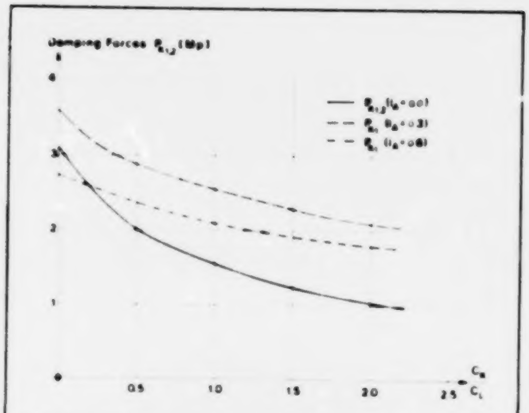
633/73
Br

Figure 14

Figure 15 shows the calculated result for the force of the damper near the impact point.

As it was to be expected, at a centered impact the increasing coupling effect induces a significant decrease of the damping forces; the coupling effect reduces the translatory damping value of the system.

If the impact point is located between the shock absorbers (example l_A = 0.3 m), as per the existant



Influence of the Connecting Coefficient on the Damping Forces for Centered and Eccentric Shocks

634 73
Br

Figure 15

calculation, the damping force of the shock absorber near the impact point will be greater than for a centered impact; if, however, the impact arrives outboard of the shock absorbers (example l_A = 0.6 m), this force will be less significant.

However, on the basis of this simplified computation we cannot yet be sure that this effect appears also in real collisions.

Perhaps we could call it our most important result that the excessive damping force which is applied to one shock absorber by an oblique impact can be reduced already at a comparatively low degree of coupling.

To avoid having this coupling effect weaken the damping at centered impacts, the throttle connections can be equipped with spring-forced valves which will become active only if too great a force for a given body stiffness arises at an oblique impact.

THE FEDERAL REPUBLIC OF GERMANY TECHNICAL PRESENTATION

SECTION TWO

PART 2

THE VOLKSWAGEN EXPERIMENTAL SAFETY VEHICLE

DR. HANS-PETER WILLUMEIT,
Department Manager
Vehicle Testing Safety, Volkswagenwerk AG

INTRODUCTION

Volkswagen began its work on the Experimental Safety Vehicle in the autumn of 1970. Eighty engineers and technicians were employed on the research and development work. In addition, a large number of craftsmen worked in the production of components, test frames and finally in the manufacture of the vehicle which we call the ESVW.

To obtain reference values during the ESVW development, the VW engineers modified two standard vehicles in our line of products, an Audi 100 with front engine and front-wheel drive and a VW 411 with rear engine and rear-wheel drive. These two vehicles, greatly modified, were subjected to various handling and accident-avoidance tests. The modified vehicles simulated the ESVW in weight distribution, over-all dimensions, engine output, axle ratios and weight.

A long series of tests, 14 destructive and 30 non-destructive was conducted with the collision test frames, which have force-distortion characteristics similar to the ESVW. Another forty crash tests were carried out with standard vehicles modified by the installation of ESV components.

In the development of the ESVW restraint system, 420 sled tests were conducted to assess the efficiency of airbags and a variety of safety belts. In the early development state, the engineers tested various head-restraint devices. One of these was a drop belt which restrained the upper torso while a drop net supported the head.

After these tests, our development team finally chose a passive combination shoulder and knee belt, similar to that shown in Figure 1a.

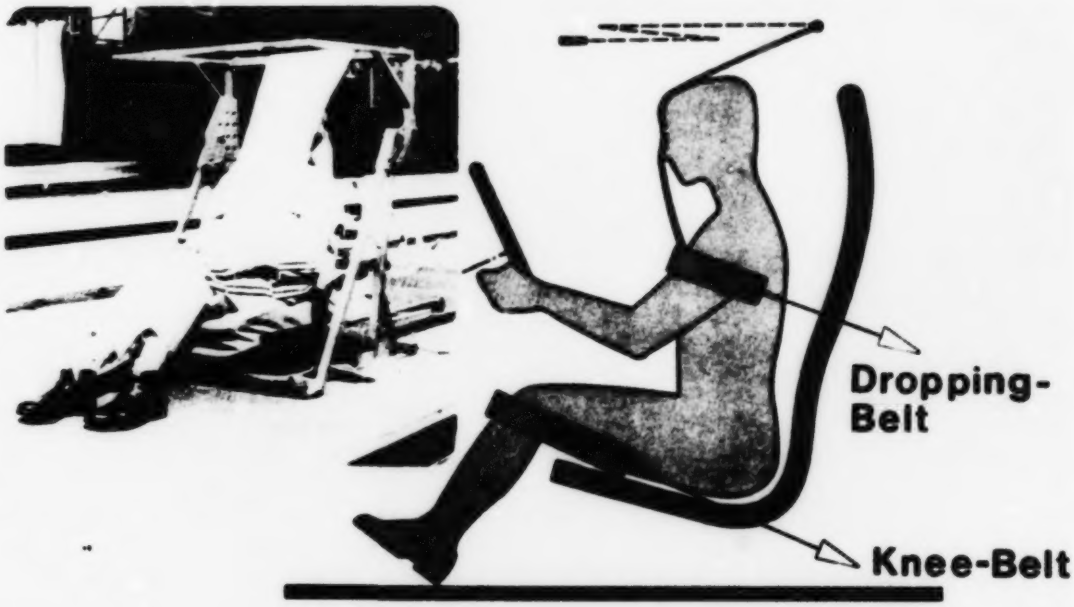


Figure 1a

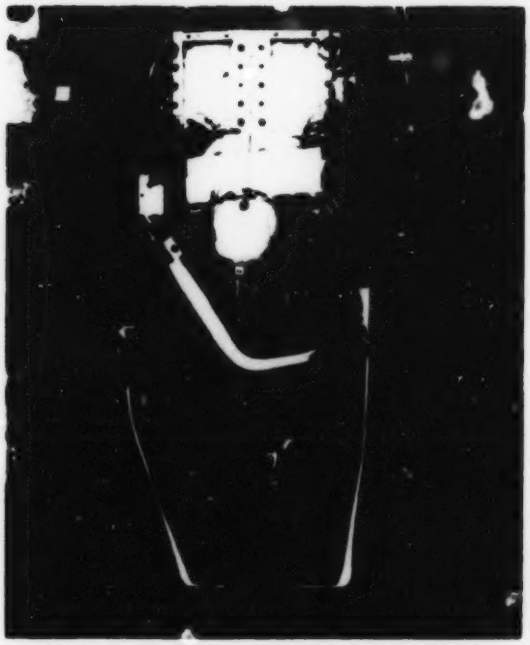


Figure 1b

Figure 1c

Numerous crash tests against both poles and barriers were conducted in the development of the high energy absorbing bumpers. All this work, both theoretical and practical, has resulted in Volkswagen's experimental safety vehicle, the ESVW, the subject of this report.

Early in the development, it became obvious that full compliance with the ESV specifications would require "exotic" materials with a high strength/weight ratio, if the vehicle weight was to be kept below the 2,000 pound (907 kg) limit. Such materials would result in much higher costs both for material and labor. It was decided not to take the unrealistic way but, instead, to build a vehicle that should be as practical as possible so that what was learned would be more useful in the real world. As a result, the ESVW is a relatively reasonably priced vehicle of high quality, in conformity with the VW corporate concept.

The Volkswagen Experimental Safety Vehicle, in appearance and size, is in line with the present standard automobile. It does not look out of place in present-day traffic (Figures 2a and 2b). It was shown to the public for the first time in May 1972 at Transpo 72, near Washington, D.C.

VEHICLE DESCRIPTION AND TEST RESULTS

The ESVW is a four-seater, four-door sedan with rear engine and rear-wheel drive.

Engine:	2.0 liter; 4 cylinder; 100 DIN HP
Transmission:	VW-automatic

Vehicle dimensions [mm]:

Length:	4,730
Width:	1,780
Height:	1,390
Wheelbase:	2,800
Track:	front 1,540, rear 1,480
Tires:	185/70 SR 14
Unladen weight:	1,464 kp

BRAKING SYSTEM

The hydraulic brake system is a dual circuit, 4 wheel disc brake system. The handbrake operates the brake discs of the rear wheels. To increase the active safety, an anti-skid system with a hydraulic brake servo (System Teves) has been built in.

Technical Data

Dual circuit operates as follows:
Circuit 1: front axle
Circuit 2: rear axle

Front Wheels

Floating frame disc brakes in wheel in front of the axle with inner-ventilated disc.
Brake pads contain electrical contacts for wear indication.

Figure 2a

Figure 2b

Rear Wheels

Floating frame disc brakes in wheel behind the axle. Solid discs. Brake pads contain electrical contacts for wear indication.

Master Brake Cylinder

Hydraulic, dual-circuit servo (master cylinder)

Handbrake

Operates the floating frame brakes of the rear wheels.

Anti-skid System

The Teves anti-skid system consists of these components (Figure 3):

1. The sensor which measures the wheel speed.
2. The electronic module that evaluates the speed-sensor reading.
3. The hydraulic control that regulates the amount of brake pressure.
4. Hydraulic pump and control valves.

After a series of tests using various control arrangements and a careful study of benefit-cost

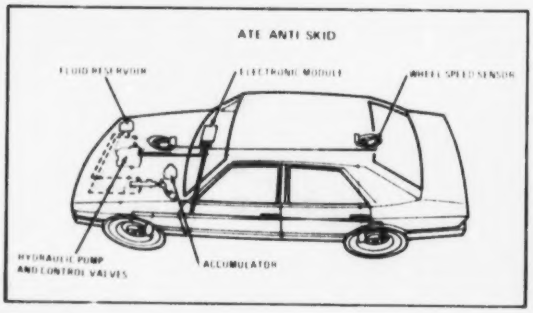

ATE ANTI SKID
FLUID RESERVOIR
ELECTRONIC MODULE
WHEEL SPEED SENSOR
HYDRAULIC PUMP
AND CONTROL VALVES
ACCUMULATOR

Figure 3

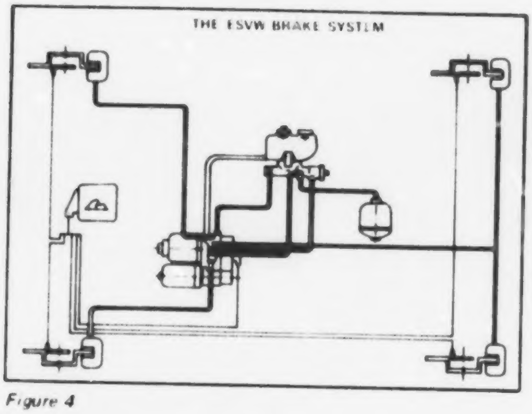


Figure 4

factors, the following control system was selected (Figure 4).

Front axle: Individual wheel control.

Rear axle: Axle control in accordance with the principle of low friction value (selected low).

To indicate a defect in the brake system, the following is built in:

Warning indicator of low fluid level in the brake fluid reservoir.

Warning indicator of applied hand brake, also serves as function check of the warning lamp.

Pad wear indicator for all brake pads.

Warning indicators in anti-skid system:

1. Tell-tale device for sensors.
2. Tell-tale device for electronic controls.
3. Tell-tale device for hydraulic controls.
4. Tell-tale device for energy supply.

BRAKING PERFORMANCE

Service Brake

U.S.-ESV (Experimental Safety Vehicle) Specifications. [1]

From a speed of 60 mph (96.6 km/h), the vehicle shall be capable of decelerating to a stop in a distance of 155 feet (47.5 m, brake retardation 7.6 m/s^2), without wheel lockup. The vehicle must remain in a 12-foot wide (3.66 m) lane. The skid number of the road surface should be SN = 70-80. The vehicle must be carrying 40% of its payload.

Under the test conditions given, the pedal force must be within the tolerance field marked in Figure 5.

The preferred maximum pedal force is 85 pounds (38.6 kp).

VDA-ESP (Experimental Sicherheits Personenkraftwagen) Specifications. [2]

The vehicle, carrying 40% of its rated load, should from a speed of 100 km/h upon application of a pedal force of 40 kp reach 75% of the mean deceleration on a 3.50 m wide lane (skid number 75 + 5) without the wheels sliding.

A definition of mean deceleration has not been given by VDA.

In order to correspond approximately with the U.S.-ESV Specifications, it is thought that by mean deceleration the normal deceleration is meant fa.

$$fa = \frac{bm \cdot 100}{g} (\%)$$

For a 75% deceleration a mean retardation of $b_m = 7.36$ m/s^2 results from this.

Results

The measured values are shown in Figure 5 and also in Table 1.

The U.S.-ESV as well as the VDA-ESP Specifications are fulfilled. However, the pedal forces are on the limit of the U.S.-ESV Specifications.

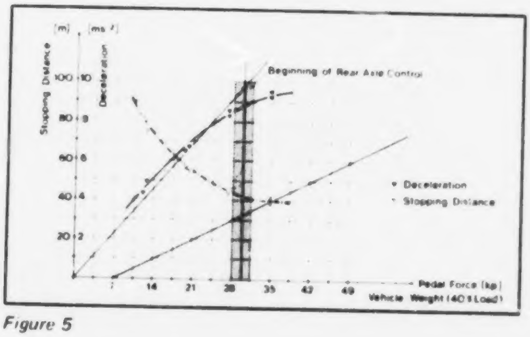


Figure 5

[illegible] (Service brake)

Service brake	US-ESV Specifications	VDA-ESP Specifications	ESVW values
Skid number	70-80	75 ± 5	[illegible]
Load	40	40	[illegible]
Lane width	3.66 m	3.5 m	[illegible]
Deceleration speed	v_0 = 96.6 km/h	v_0 = 100 km/h	[illegible]
Pedal force	less than 38.6 kp	less than 40 kp	28.6 kp start of brake pressure regulation greater than 21 kp
Brake retardation	greater than [illegible]	greater than [illegible]	[illegible]
Stopping distance without locking of one wheel	47.5 m	none	48 m

Table 1

Fade Tests

U.S.-ESV Specifications

Conditions and nominal values in accordance with SAE Recommended Practice J843 a (June 1966).

VDA-ESP Specifications

None.

Results

The VW standard production vehicles already comply with the fade tests. A measurement therefore was not considered necessary.

Load-dependent Brake Performance

U.S.-ESV Specifications

Under all load conditions, the deceleration should be 80% of the braking effect that the vehicle has when it carries 40% of its payload (i.e. the stopping distance under all load conditions may be 1.25 times longer than the stopping distance with 40% of the payload). These conditions shall be valid under all coefficients of friction $\mu = 0.2 - 0.8$.

The determination of the stopping distance is carried out in accordance with the requirements given in **Service Brake**.

VDA-ESP Specifications

The vehicle, empty or loaded, with coefficient of friction from $\mu_{max} = 0.2 - 0.8$ must reach a degree of efficiency of at least 0.75.

The degree of efficiency is the relationship of the deceleration obtained to the maximum possible value with the appropriate coefficient of friction μ_{max} and ideal brake power distribution.

$$fa' = \frac{b \cdot 100}{\mu_{max} \cdot g}$$

Results

The measured results, shown in Figure 6 of fa' = 75% represent a more severe condition than that in the U.S.-ESV Specifications. Only the degree of efficiency fa' was determined, and that with 100% payload as this represents the severest condition.

U.S.-ESV and VDA-ESP Specifications have been fulfilled.

Parking Brake

U.S.-ESV Specifications

The vehicle, when fully loaded, using a friction type mechanical brake shall be capable of holding on a 30% grade in the forward or reverse directions. Actuation effort shall be less than 90 pounds (41 kp) for a hand system and 125 pounds (57 kp) for a foot system.

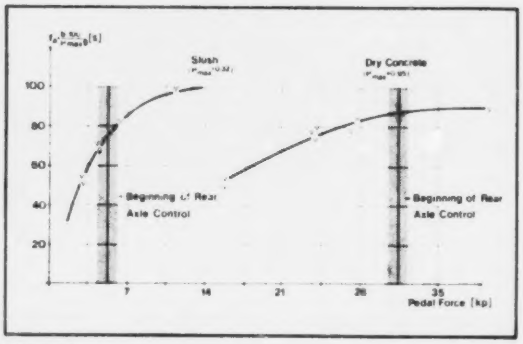


Figure 6

VDA-ESP Specifications

The parking brake is to be designed as a mechanical friction type brake. It must be capable of holding a fully loaded vehicle on a 30% grade in both directions. The actuating effort of 40 kp for hand operation and 60 kp for foot operation must not be exceeded.

Results

See Table 2.

The U.S.-ESV as well as the VDA-ESP Specifications are fulfilled.

Comparison of the measured values with the US-ESV and VDA-ESP specifications (Parking brake)

Parking brake	US-ESV	VDA-ESP	ESVW - values
hand brake			
[illegible]	[illegible]	[illegible]	
grade (in both directions)	30%	30%	
manual force	less than 41 kp	less than 40 kp	manual force forwards 21.5 kp reverse [illegible] kp
foot brake			
foot force	less than 57 kp	less than 60 kp	N.A. as the ESVW is equipped with a hand parking brake

Table 2

Emergency Brake

U.S.-ESV Specifications

In the event of a booster failure or leakage in one circuit of the system, the remaining portion of the service brake system shall continue to operate. The vehicle stopping distance with 40% payload and a skid number of 70 - 80 shall not exceed 343 feet (104 m, 0.35 g retardation) if from a speed of 60 mph (96.6 km/h) the brakes are actuated with a pedal force of not more than 150 pounds. In the event of front brake circuit failure, the pedal force should lie within the tolerance field shown in Figure 7.

VDA-ESP Specifications

None

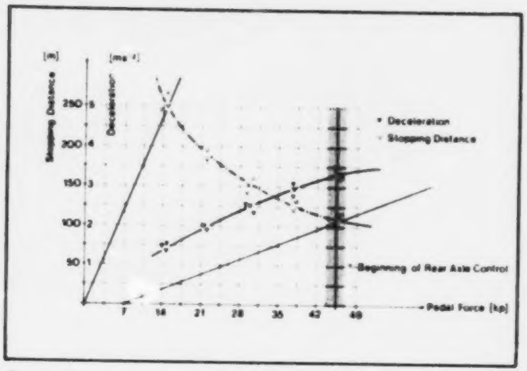


Figure 7

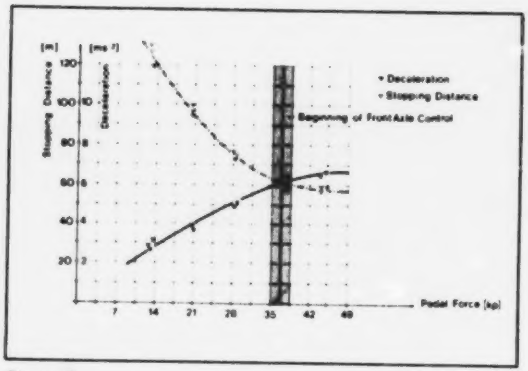


Figure 8

Comparison of the measured values with the
US-ESV and VDA-ESP specifications (Emergency brake)

Emergency brake	US-ESV	VDA-ESP	ESVW Values
failure of the rear circuit foot force	less than 68 kp	—	
stopping distance	less than 104 m	—	foot force 45 kp b = 0.65 g, stopping distance 56 m
failure of the front circuit			foot force 45 kp b = 0.35 g, stopping distance 103.5 m

Table 3

Results

Figure 7 shows the relationship of the stopping distance or the brake retardation to the pedal force with a failure of the front-wheel brake circuit.

Shown in Figure 8 is the curve of the stopping distance and the brake retardation with a failure of the rear-wheel brake circuit.

Table 3 lists the principal measuring points. The U.S.-ESV Specifications are fulfilled.

TIRES AND WHEELS

To reduce the unsprung masses, an aluminum cast alloy was selected for the slotted wheels 5½ J x 14 H2 (Figure 9). Centering on the hub guarantees perfectly smooth wheel rotation. Four mounting bolts are used on each wheel.

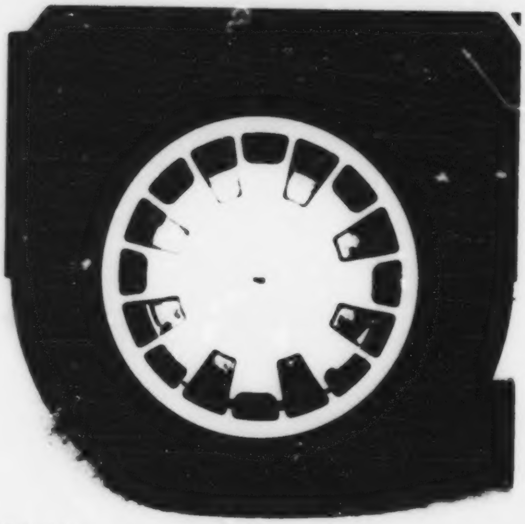


Figure 9

To give improved riding comfort, textile radial tires with tubes are used (tubeless tires can also be used). The wheel bolts are covered by a plastic wheel cap, the retaining ring of which engages in the center of the wheel.

The spare wheel is mounted on a pivoting bracket under the front luggage compartment.

AXLES AND STEERING

Front Axle

Due to the space restriction resulting from the use of hydraulic energy absorbers, a McPherson suspension strut axle with coil springs was used (Figure 10).

Axle data (fully laden) 340 kp payload

Track	1,540 mm
Camber	0°
King pin inclination	12.4°
Toe	0°
Caster	2.1°; 30 mm
Steering rolling radius	10 mm negative
Wheel lock, inner wheel	38.8°
Wheel lock, outer wheel	30.1°
Turning circle dia	11.4 m
Roll center	284 mm
Damper inclination	6°

Front Suspension

The suspension strut is pivot-mounted at the top in a bonded rubber mounting and a ball bearing. The lower mounting is by a ball joint to the control arm which consists of two U profiles.

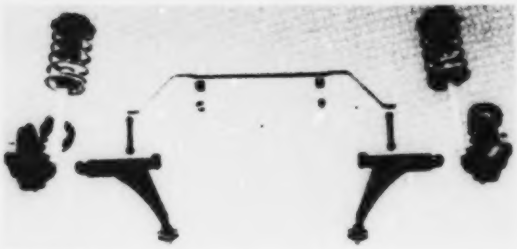


Figure 10

The control arm is attached to the body by a bracket with two rubber mountings. The control arm mounting is outside the crunch zone. All bearings are maintenance-free (Figure 11).

The wheel hub, which is separate from the vented brake disc, is mounted by means of two tapered roller bearings on the spindle of the forged steering knuckle. Bearing play is adjustable. The brake caliper and the brake anti-skid sensor are also attached to the steering knuckle (Figure 12). The steering knuckle is bolted to the suspension strut. One of the two bolts is eccentric and is used for camber adjustment.

Front suspension is by coil spring with a linear and an auxiliary spring with progressive properties. It is unitized with the double-acting hydraulic shock absorber. When selecting the coil springs, a body frequency of .1 Hz for the front axle was taken as the basic figure. (According to the ESV specifications: 0.9 to 1.1 Hz at 40% payload without shock absorber). Taking the mean ratio of 1.07 and the auxiliary spring rate into consideration, this results in a coil spring of c = 1.44 kp/mm.

A stabilizer with a diameter of 19 mm, c = 6.71 kp/mm prevents undue body roll. The stabilizer is attached to the body by two rubber mountings and to the control arms by a rubber-mounted lever on each side.

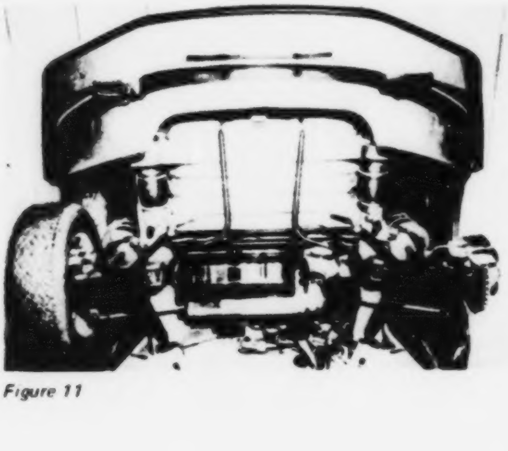


Figure 11

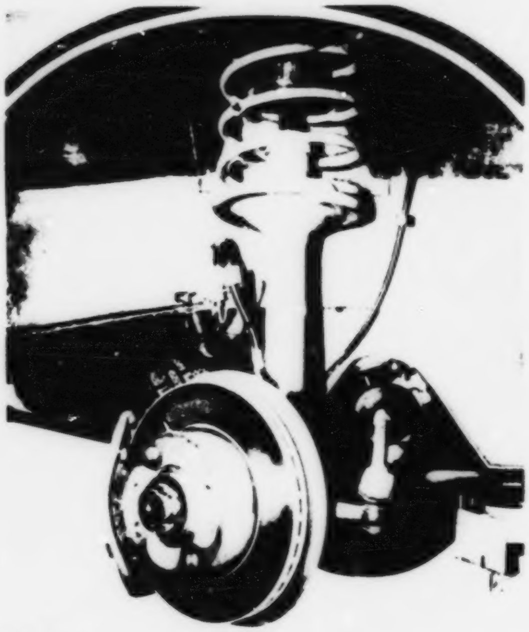


Figure 12

Rear Axle

Double-jointed trailing arm axle with spring damper and stabilizer (Figures 13, 14, and 15).

Axle data (fully laden) 340 kp payload

Track	1,480 mm
Camber	2° ± 10'
Toe	10' + 10'
Roll center	38 mm
Trailing arm axis	8°

Rear-axle Support

The rear-axle support comprising trailing arm mountings and a welded-on transmission stop, consists of a closed-U profile and is rigidly bolted to the body side member on both sides. The inner mountings permit toe-in and camber adjustment. The axle support is designed so that in a rear-end impact it stops the engine transmission unit from being pushed forward and deforming the passenger cell.

Rear Suspension

The trailing arm consists primarily of two shells welded together to form a profile, a bearing tube for the location of the two maintenance-free rubber mountings for attaching the trailing arm to the rear-axle support, and the wheel bearing housing. The brake caliper, containing the handbrake as well as the anti-skid sensor, is attached to the wheel bearing housing.

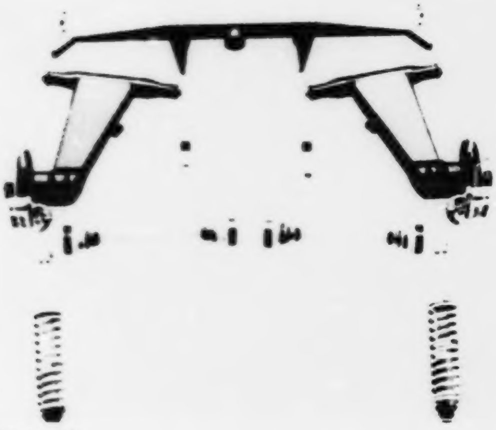


Figure 13

The wheel hub – separate from the brake disc – is mounted in the wheel-bearing housing by means of a self-sealing ball bearing. There is an oil seal fitted on each side.

Two joints shafts of different lengths, resulting from the off-center location of the differential, actuate the rear-wheel shafts. Two constant-velocity joints on both shafts offset the angle differences resulting from the compression and rebound action of the suspension. Suspension is by cylindrical coil springs with linear and rubber auxiliary springs with progressive properties.

Both are attached to double-acting hydraulic shock absorbers to form a unit. A body frequency of 1.21 Hz at the rear axle was the basis when selecting the coil spring (according to ESV specification, 1.2 to 1.4 Hz with 40% payload without shock absorbers).

The auxiliary spring rate and the mean ratio of 1.06 result in a coil spring rate of 2.46 kp/mm.

Body roll is reduced by a stabilizer, 17 mm dia., c = 2.37 kp/mm which is attached by two rubber mountings on each side to the rear-axle support and to the trailing arms.

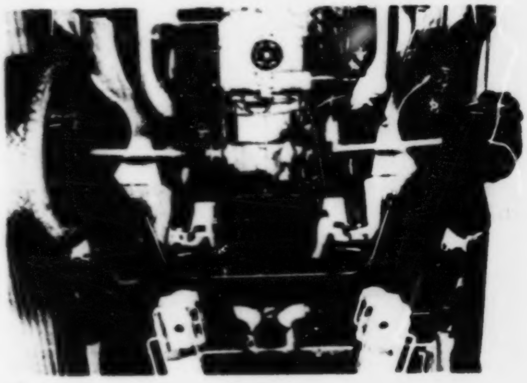


Figure 14

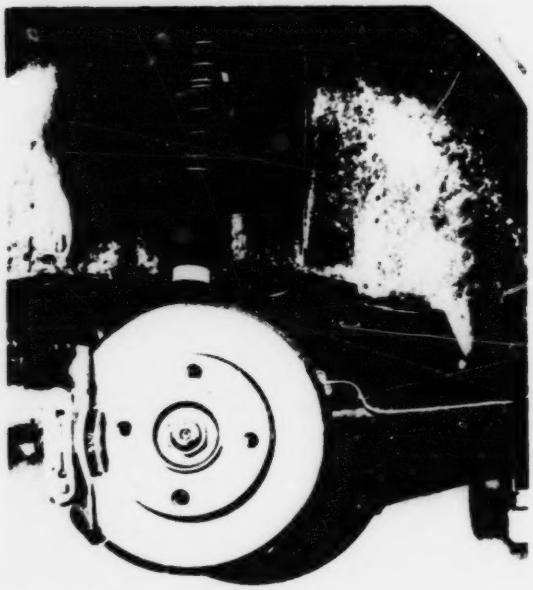


Figure 15

Steering—Adjustment Of Steering Wheel And Pedals

Steering is by a maintenance-free, self-adjusting rack-and-pinion steering gear with symmetrical tie rods, steering damper, steering control (co-pilot) and an adjustable steering column. The rack-and-pinion steering-gear, with helical toothing and self-adjusting, two-piece, prism-located rack, represents an optimum with regard to easy movement, efficiency and freedom from road-induced-shock. Technical features are shown in Figure 16.

The steering column, together with the pedal cluster, is adjustable to fit the various body sizes of drivers. This is necessary because for safety reasons the seats in the ESVW are fixed, forming structural elements of the body.

To determine the necessary range of adjustments, a seating buck was built, containing (model ESVW, scale 1:1) a pedal cluster adjustable in the vertical and horizontal planes, a steering column separately adjustable in the same planes.

For the adjustment range, the following values were given:

Pedal cluster:	horizontal:	200 to 270 mm
	vertical:	60 to 100 mm
Steering column:	horizontal:	140 to 165 mm
	vertical:	110 to 185 mm

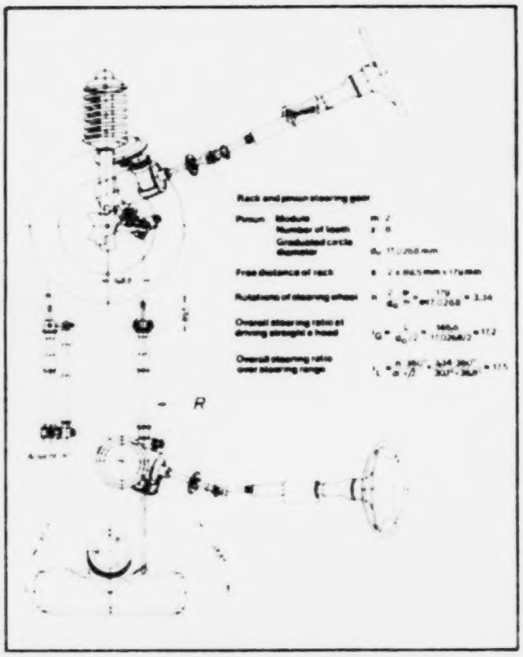

Rack and pinion steering gear
Pinion
Module
Number of teeth
Graduated circle diameter
Free distance of rack
Rotations of steering wheel
Overall steering ratio at driving straight a head
Overall steering ratio over steering range
R

Figure 16

During the research, 30 test persons were used to determine the best location of the pedal cluster and steering column for maximum comfort. Six of the test persons were female and 24 male. Of the females, three were 5-50% women, two were 50-95% women, and one was an over 95% woman. Of the 24 males, eight were 5-50% men, twelve were 50-95% men, and four were over 95% men. From these 30 test measurements it was established that the pedal adjustment range in the horizontal direction was not sufficient for the 5% woman.

For the construction and installation of the adjustable pedal cluster the horizontal adjustment travel was limited to 200 mm because of the limited interior space available, giving for men larger than 95% a very slight reduction in comfort. A common adjustment of pedal cluster and steering column was selected in order to retain operating simplicity.

STEERING TESTS

Data Of Test Vehicle

The test vehicle was a standard production vehicle converted to ESVW running gear specifications. The dimensions of the converted vehicle are:

Wheelbase:	2,800 mm
Front track:	1,540 mm
Rear track:	1,480 mm
Overall length:	4,730 mm

When no other specifications are given, the vehicle weight for all tests is equal to the unladen weight of the ESVW (1,464 kp) plus 27% of the load, assuming a maximum load of 340 kp.

The weight distribution is:

Front axle load:	716kp
Rear axle load:	840 kp
Test weight:	1,556 kp
Tires:	Michelin XAS 180 HR 15
Tire pressures:	Front/rear 2.0/2.2 kp/cm^2
Engine output:	100 DIN HP
Steering ratio:	$i_m = 21.7:1$

The ESVW has a steering assist device (silent co-pilot) that adds steering input only when pavement irregularities, side winds, roll, etc., make such input corrections necessary (Figure 17). When such exterior interferences make added input necessary, a steering-angle correction is made by the co-pilot to keep the vehicle from reacting to the interference. The maximum correction corresponds to an angle of ± 65° at the steering wheel.

During development of the ESVW, the steering characteristics of the vehicle could be varied within a very wide range with the help of the co-pilot. This device turned out to be an ideal development aid in designing the best possible steering characteristics for the ESVW.

Steering is defined by the U.S.-ESV and the VDA-ESP Specifications as (1) the steady state yaw response, (2) the steady state yaw response with different lateral accelerations, with different power applications and changes in loading, (3) the transient yaw response and (4) the steering returnability when the steering wheel is released.

Steady State Yaw Response

The U.S.-ESV Specifications state the following in respect of steady state yaw response:

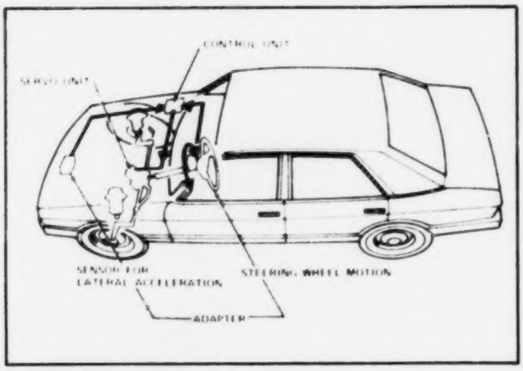

CONTROL UNIT
SERVO UNIT
SENSOR FOR LATERAL ACCELERATION
STEERING WHEEL MOTION
ADAPTER

Figure 17

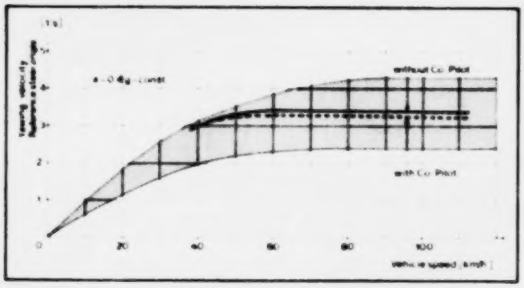


Figure 18

In response to a steering input for which the lateral acceleration is 0.4 g the vehicle must:

Maintain a steady state yaw response at forward velocities of 40 km/h (25 mph), 80 km/h (50 mph) and 112 km/h (70 mph) within the envelope defined in Figure 18.

The VDA-ESP Specifications do not stipulate any special yaw response but merely require that the obtainable steady state lateral acceleration shall be not less than 0.65 g with a test circle diameter of 80 m. Figure 18 shows the steady state yawing velocity versus the vehicle tangential velocity. In this diagram the vehicle yaw angle velocity based on the reference front wheel angle is entered on the ordinate.

As can be seen in Figure 18 the U.S.-ESV Specifications are adequately fulfilled without and with the co-pilot.

Steady State Yaw Response With Different Lateral Accelerations And Various Loads

The U. S.-ESV Specifications state:

When the lateral accelerations change, the response curves shall have the same characteristic shape as shown in Figure 18 for 0.4 g, which means that the yaw response should not change to any degree. Furthermore, in relation to the 0.4 g yaw response curve in Figure 18, as lateral acceleration increases and at a constant velocity the yaw response curve shall progressively move downwards, as lateral acceleration decreases from 0.4 g the yaw response curve shall progressively move upwards. The VDA-ESP specifications require additionally that a change of the loading shall not affect the steady state yaw response.

Test results have shown that when the lateral accelerations change, the shape of the curves in Figure 19 (without co-pilot) and Figure 20 (with co-pilot) is the same as it is with a lateral acceleration of 0.4 g. When the co-pilot is switched on a parallel displacement of the curves takes place (compare curves for same lateral acceleration in Figures 19 and 20).

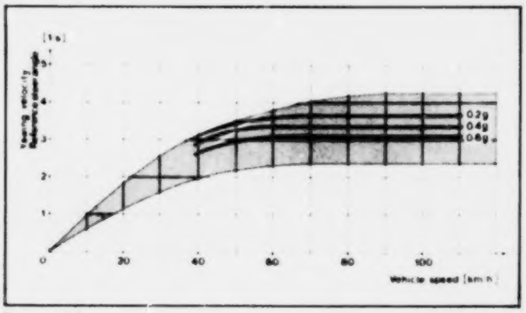


Figure 19

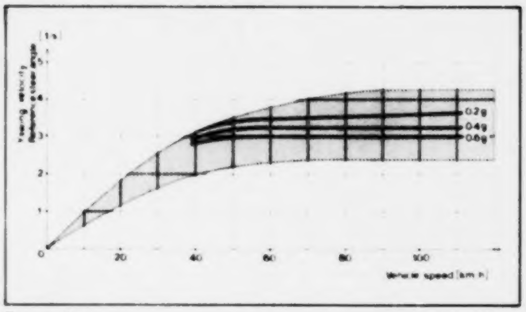


Figure 20

The U.S.-ESV Specifications require that in relation to the 0.4 g curve the yaw response curve shall move progressively downwards as the lateral acceleration increases and that the yaw response curve shall move progressively upwards as the lateral acceleration decreases is a universal condition that is fulfilled to all practical purposes by conventional vehicles. The tests for the steady state yaw response with changes in the loading were carried out with an additional load of 100 kp at the front and without the co-pilot. Figure 21 shows the influence of the changing load. The effect of an additional load at the front (which occurs only on cars with forward luggage compartments) remained within acceptable limits.

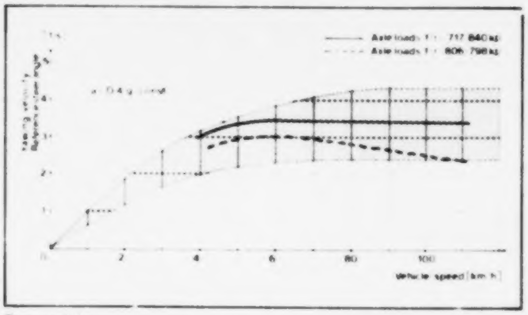


Figure 21

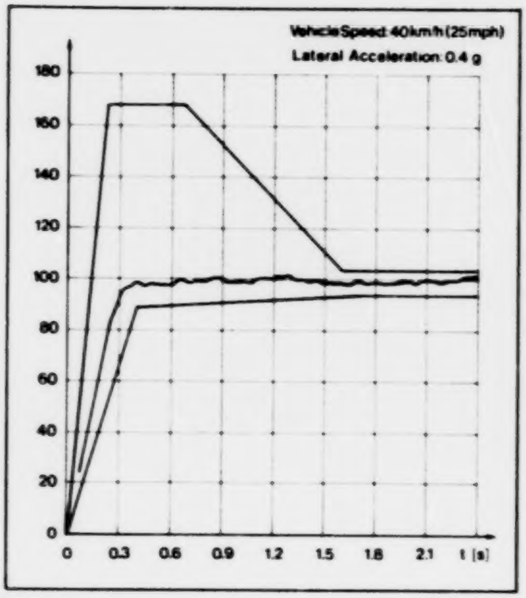


Figure 22

Transient Yaw Response

The U.S.-ESV Specifications for transient yaw response are as follows:

With a steering input applied at a rate no less than 500°/sec and held constant at a value which produces 0.4 g steady state lateral acceleration with test velocities of 40 km/h (25 mph) and 112 km/h (70 mph), the transient yaw response shall be within the envelope described in Figures 22 and 23.

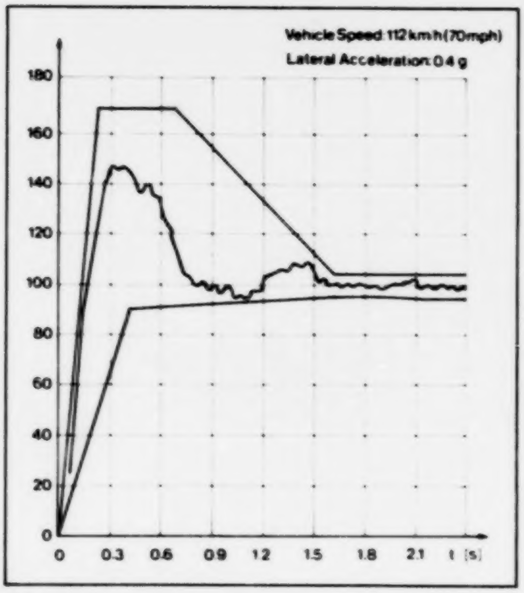


Figure 23

The VDA-ESP Specifications contain no details in respect of transient yaw response.

The Figures 22 and 23 also show that the requirements are fulfilled by the vehicle. The steering control (co-pilot) has no influence here on the handling.

Returnability

The U.S.-ESV Specifications for steering returnability are as follows:

The vehicle shall be driven at 40 km/h (25 mph) and 80 km/h (50 mph) around a fixed radius sized to achieve 0.4 g. At a predetermined point of the vehicle path, the driver shall release the wheel completely and the vehicle must:

Not have a yaw rate greater than 4 degrees per second at any time, greater than 2.0 seconds after steering wheel release with a test velocity of 80 km/h (50 mph). From a test speed of 40 km/h (25 mph), the yaw rate 2 seconds after steering wheel release must be zero. The requirement for steering returnability stability is that the relative yaw response versus time must be within the envelope shown in Figures 24 to 27. These figures also show that the vehicle, with and without co-pilot, fulfills the requirements.

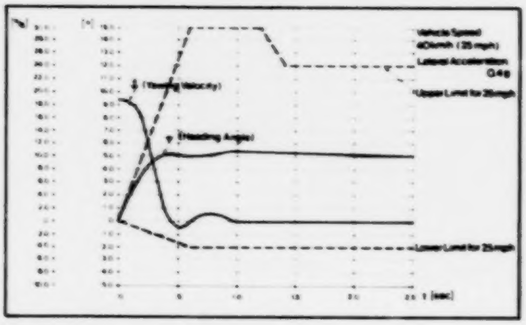


Figure 24

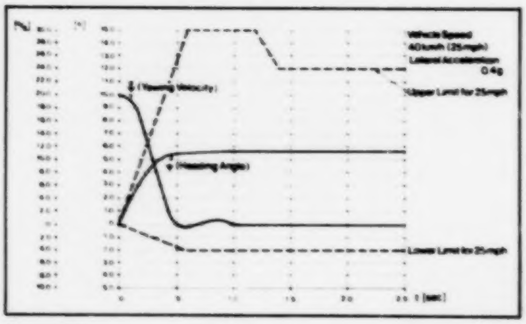


Figure 25

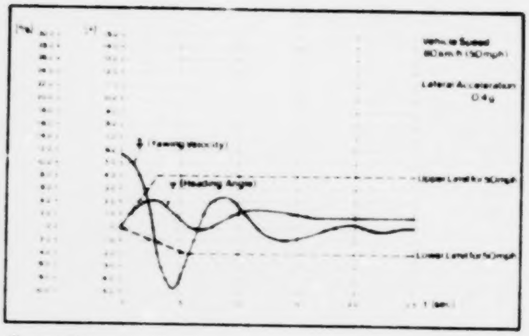


Figure 26

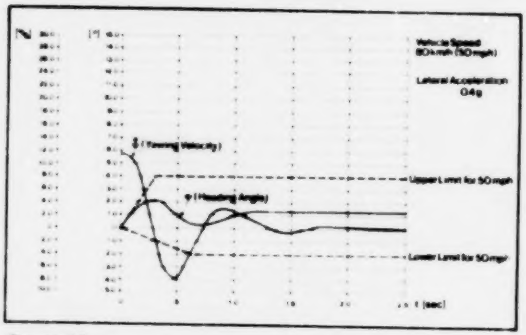


Figure 27

HANDLING

Lateral Acceleration

To test the lateral acceleration, the U.S.-ESV Specifications state that the vehicle, while being driven on a 100 foot radius skid pad at a constant velocity, shall remain stable as shown in Table 4 of the Specifications. The certification circle shall be completed with a speed variation of ± 1 mph. The required "g" load when traveling round the circle must be obtained with a total steering wheel correction movement of less than 10°.

The VDA-ESP Specifications state that the results must not be below the values given in Table 4 and the diameter of the test circle shall be 60 m. The test speed for each lateral acceleration test and for each complete round must be maintained to within ± 2 km/h. The steering correction required must not exceed 10° total steering wheel movement.

Tests for the lateral acceleration as specified in Table 4 show that with or without co-pilot there is no difference in the lateral acceleration achieved. The required lateral accelerations were fulfilled with and without steering corrections.

The ESVW fulfilled the requirement that the lateral accelerations on a wet surface shall be in the same relationship to those achieved on a dry surface as with the skid numbers or adhesion friction coefficients for dry and wet roads.

As shown in Table 4 the differences in the lateral accelerations obtained with wet roads and with dry roads are not very large. In tests with nominal tire pressures the lateral accelerations obtained on wet roads are only about 5% lower than on dry roads.

Control At Breakaway

The U.S.-ESV Specifications suggest the following test for this condition:

The vehicle shall be operated on a skid pad, in a given circular path, at maximum lateral acceleration until a steady state condition exists. The vehicle speed shall then be gradually increased until any point on the vehicle center line within the wheel base has moved radially outward a distance of 10 feet (3.048 m) from the initial path. At this time, the throttle will be closed, initiating the test of control at breakaway. Measuring time from the instant the throttle closes, the vehicle must be capable of regaining and following the initial path ± 2 feet (± 0.61 m) within no more than 4 seconds.

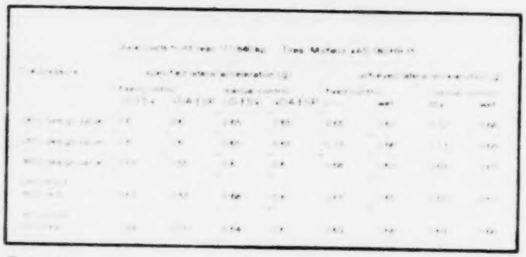


Table 4

This test will be conducted at a steady state circular path of 100 feet (30.48 m) and 225 feet (68.5 m) radii. Further conditions to be adhered to when conducting these tests are as follows:

No braking shall be performed on the vehicle during recovery.

Steering wheel movements necessary to regain the original path must not exceed 500° applied at a rate not to exceed 500°/s.

The VDA-ESP Specifications do not give any details on this test. The times obtained in the tests, which were only performed without co-pilot due to lack of time, are given in the following table:

	Radius	Time achieved
Standard vehicle	100 ft.	2.3 seconds
With ESVW running gear	225 ft.	3.0 seconds

Directional Stability

According to the U.S.-ESV Specifications, this describes the stability of a vehicle when subjected to cross-winds or pavement irregularities.

This concerns not only the course deviation but also the steering force required to compensate for small influences and the steering force required when power assistance fails.

Crosswind Sensitivity

The U.S.-ESV Specifications laid down the following tests:

The test vehicle shall be subjected to a step function crosswind application of 80 km/h (50 mph) ± km/h (± 5 mph). Total course deviation for a 20-foot exposure shall be measured two seconds after initiating test and shall be less than the values shown in Figure 28. The test shall be carried out at speeds of 48 km/h (30 mph), 80 km/h (50 mph) and 112 km/h (70 mph) on a level road surface with head or tail wind velocities of less than 8 km/h (5 mph).

The VDA-ESP Specifications in respect of crosswind sensitivity state:

The course deviation shall not exceed the values given in Figure 28.

The crosswind velocity shall be 80 ± 10 km/h (50 ± 6.25 mph) and the length of the exposure shall be 6 m. The vehicle speeds specified are 50, 80 and 110 km/h.

The tests are made on a level surface with a maximum head or tail wind of < 10 km/h, with steering locked. The total course deviation is measured two seconds after commencement of the test.

The test results, with and without co-pilot, are shown in Figure 28.

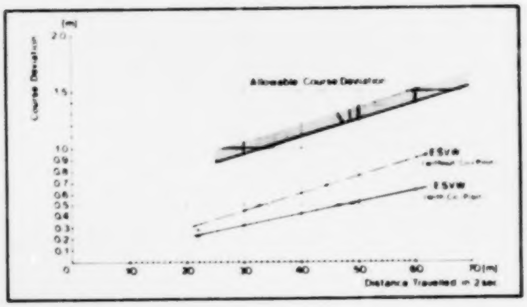


Figure 28

Pavement Irregularity Sensitivity

U.S.-ESV Specifications states:

A ridge one inch high and with a radius of one inch is provided on the road surface. This ridge shall lie along a line intersecting the directional vector of the vehicle at an angle of 30 degrees.

Total allowable course deviation is one foot, measured two seconds after driving over the ridge at speeds between 48 km/h (30 mph) and 112 km/h (70 mph). VDA-ESP Specifications contain no detail for this test. The signal given when driving over the specified ridge could not be evaluated because the effect on the vehicle was so slight.

No course deviation could be established.

Steering Control Sensitivity

U.S.-ESV Specifications state that the steering torque required to cause a yaw rate of two degrees per second shall exceed five inch pound (5.77 cm kp) when applied steadily at the rim of the standard wheel. These conditions are for speeds of 48 km/h (30 mph), 80 km/h (50 mph), and 112 km/h (70 mph).

VDA-ESP Specifications state that the torque applied steadily at the rim of the steering wheel to cause a yaw rate of two degrees per second at a speed of 50 km/h shall exceed 6 cm kp.

The following test values were obtained:

Vehicle speed [km/h]	Steering wheel torque [kp cm]
50	12.0
80	12.5
110	13.0

Overturning Immunity

To demonstrate overturning immunity, two tests, the J-turn and the pylon course, are specified.

J-turn

The U.S.-ESV Specifications for a 180° turn represent the most severe dynamic driving requirement:

To demonstrate overturning immunity, the vehicle shall not overturn under any combination of braking and/or steering at any velocity on a level paved track. As the minimum the vehicle must not overturn: when negotiating a J-turn at velocities of 48 km/h (30 mph), 80 km/h (50 mph), and 112 km/h (70 mph) with a steering-wheel deflection rate applied at no less than 500°/s and to a deflection value of at least 180°.

The VDA-ESP Specifications with respect to the overturning immunity of a vehicle are not so severe. They are: sudden steering-wheel deflection without reduction of power at speeds of 80 km/h, with a steering-wheel deflection rate of 500°/s and a steering-wheel deflection angle of 180°.

J-turns were made, overturning did not take place (Figure 29). The overturning immunity with any combination of steering and braking maneuvers at any velocity was not tested as it is known that for

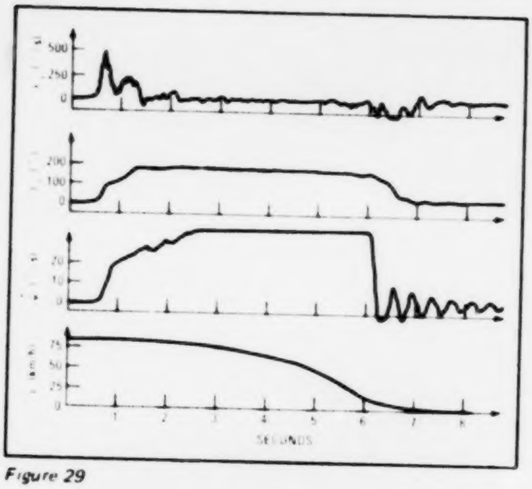


Figure 29

physical reasons vehicles of this size cannot be immune to overturning under such extreme conditions. Furthermore, the safety gain represented by the fulfillment of this test is considered to be doubtful.

Pylon Course Test

As a further test of overturning immunity, a pylon course test shall be carried out. According to U. S. - ESV Specifications, this is to be carried out at a speed of 72 km/h (45 mph) with pylons spaced 100 feet apart without the vehicle overturning. No statement is made about the number of pylons to be driven around.

The VDA-ESP Specifications state that a vehicle should not overturn when driven through a pylon course at a speed of 80 km/h (50 mph) with pylons spaced 30 m apart. No detail is given to the number of pylons to be driven around.

The U. S. - ESV and the VDA-ESP Specifications were both fulfilled. A course of 5 pylons was negotiated. The test was carried out without the co-pilot engaged.

VISIBILITY

The U. S. - ESV Specifications were used as the basis of the design of the field of view.

As the driving seat adjusts only vertically, it was possible to deviate from the eye-ellipse as described in SAE-J 941b so that a very small eye-movement field was obtained. On the basis of this eye-movement field, the following field of view was obtained (Figure 30).

Driver's horizontal eye range = 76° to front
43° to rear

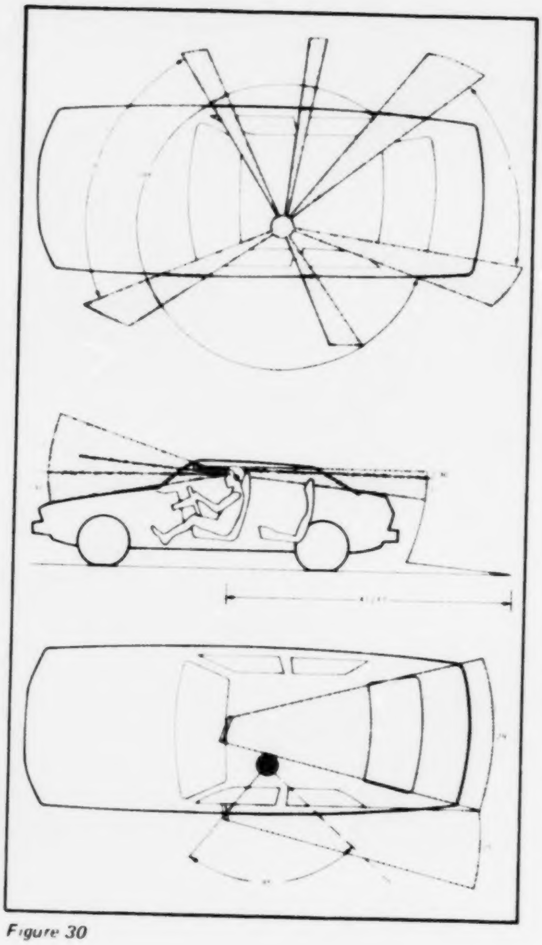


Figure 30

In the 288 are there are four body pillars. The driver's vertical eye range = 17° above the horizontal line
3° under the rear view mirror

Below the horizontal line, the angle of view is 7° 40'. When the interior rear-view mirror is used, the angle of view to the rear is 1° 30' above the horizontal line and 5° below the horizontal line.

With the interior mirror, the ground can be seen at a distance of 13.5 m behind the driver. The horizontal angle of view to the rear in the interior mirror is 29°, in the exterior mirror it is 15°.

Windshield Wipers

The windshield is cleaned by a single-arm wiper with an effective blade length of 455 mm. The pivot point of the 515 mm wiper arm is near the vehicle center. The working angle of 130° was obtained by suitable arrangement of the wiper linkage (Figure 31).

Compared with the normal two-arm wipers, this

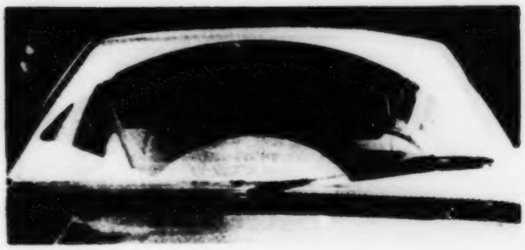


Figure 31

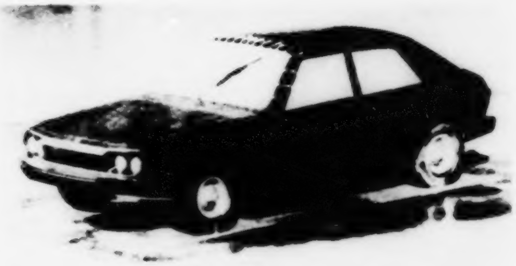


Figure 32

one-arm system has the advantage that the blade is in the most favorable aerodynamic air flow in every position (Figure 32). This reduces the likelihood of blade lift at high speeds. Furthermore, the water control along the blade rubber is very good.

The windshield wipers are controlled by moving a lever on the right of the steering column in the steering-wheel plane as on most current production vehicles (Figure 37 on page 125).

Lever position Down I	Wipers slow (also headlight wipers when lights are on)
Lever position Up I	
Lever position Up II	Wipers fast
Flick wiping	By flicking lever lightly upwards

Windshield Washer

The windshield washer is operated by an electric pump. The water for the double jet is stored in a 1.5-liter container in the front luggage compartment. The washer is controlled by lifting the wiper control lever towards the steering wheel. When the lever is in Position I, as described above, the windshield can only be sprayed when the lights are switched on and thus, in night driving, the headlights are automatically cleaned each time the windshield is cleaned. During daylight use of the windshield washer, the headlight spray does not function thus conserving water, when the lights are not in use.

Headlight Washing System

The headlight wipers have flexible blades which park at the rim of the headlights when not in use (Figure 33). The wiper shaft is located between the dual headlights. The wipers for both pairs of headlights are powered by one electric motor via Bowden cables.

The water used to clean the headlights is stored in a 1.5.l container and pumped to the separate jets for each headlight by the electric pump. The water supply is adequate for 30 cleaning cycles.

The headlight wipers work when the control lever is at Position I lower and the headlights are on. To stop the headlight wipers (with the lights on) and continue windshield wiping, the lever is moved to Position I upper (slow) or II upper (fast).

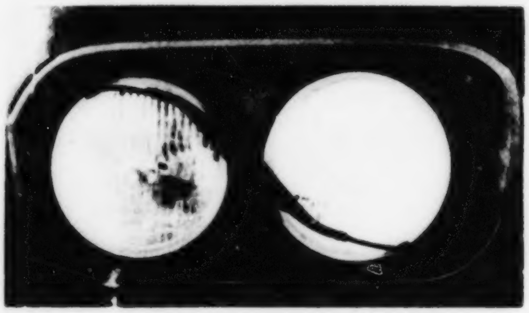


Figure 33

LIGHTING AND CONTROL SYSTEM

Dual headlights:	Halogen headlights plus side lights which can also be used as parking lights, left or right. The headlights turn off automatically when the ignition is switched off. The side lights remain on.
Foglights:	Halogen lights below the bumper.
Front turn signals:	Visible three-dimensionally (Figure 34).

The turn signals are kept clean by the air flow past the upper and side surfaces.

Rear lighting:

Each rear-light cluster includes, from bottom to top, the following:

Brake light, turn signal and an additional brake light which works as follows:

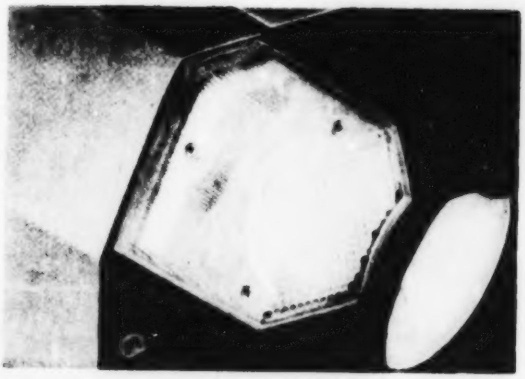


Figure 34

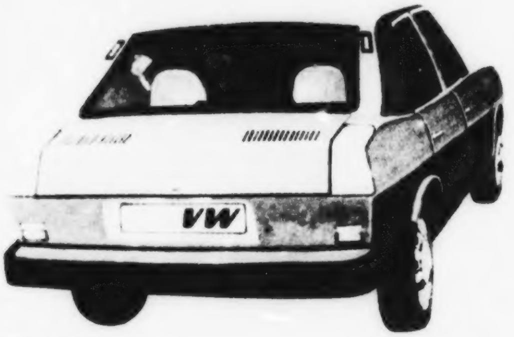


Figure 35

In the wiring circuit from the brake pedal contact switch to the brake lights is a modulator. In normal braking, the modulator supplies an impulse which initiates three or four flashes of the brake lights at short intervals. Then the brake lights come on and remain on steadily until the brake is released.

When the brakes are applied hard and a vehicle retardation of 0.4 g is obtained, the bright rapid flashes continue until the vehicle retardation drops below this value. Then the brakes remain on steadily until the brake is released.

The location of the brake lights is such that they can be seen even if another vehicle cuts in between the ESVW and the next vehicle following. Both the turn signals and brake lights are connected to the dual-intensity lighting system.

Tail lights:

Four green tail lights, located at two different distances from one another (two high on the rear pillars, two mounted low, immediately above the bumper), provide orientation points for a following driver to help him estimate the distance he is from the ESVW. Red reflectors are incorporated in the lower tail lights.

License plate lights

Two license plate lights are fixed to the rear lid to illuminate the plate from above.

Parking lights

These can be switched on separately for each side.

Dual Intensity System For Signal Lights

Day intensity: Normal current circuit.

Night intensity: When the head lights are switched on, a relay is actuated which brings a series resistance into the circuit to reduce the brightness of the signal lights.

When the foglights are switched on, the system is returned to the normal current circuit and the turn signal-lights are at full intensity.

The front and rear turn signals and the brake lights are controlled by this dual intensity system.

Emergency warning lights:

The emergency warning blinker system uses the front and rear turn signals, as is normal.

Lamp failure indicator:

Each pair of lights is controlled by a differential coil, consisting of two electrical coils working against each other. The current to one lamp of each pair flows through a separate coil. If both lamps of the pair are in order, the magnetic effect is cancelled by the identical current flowing in the two coils. If one lamp fails, the condition in the coils is unbalanced and the resultant magnetic field switches a reed contact located within the coils, causing a warning lamp to light on the instrument panel. If a turn signal lamp fails, the solenoid in the change-over relay switches in the appropriate brake light which then takes over the function of the turn signal.

Interior lights:

Interior lights are located in the roll bar in the passenger compartment. Front and rear luggage compartment lights are provided.

Diagnosis socket: For quick and accurate maintenance, a VW computer-diagnosis socket is located in the front luggage compartment.

Instrument panel:

A safety instrument panel combined with two knee bars is installed (Figure 36). The knee bars have built-in safety belts. The bars consist of a U-shaped metal stamping supported in the direction of motion by two bars fixed between the tunnel and A-pillars.

On the side facing the passenger, the knee bar is covered with a 15-mm thick layer of PU foam and a layer of vacuum-molded ABS foil. This foil covers the knee belts and has a center slit which is held together by a sliding fastener that opens in a crash to release the knee belts.

The instrument panel itself consists of a metal stamping with PU foam padding covered with ABS foil. The panel is bolted to the knee bar. Due to the integration with the elevated center tunnel, no supports are required. The cowling for the instruments is fully integrated in the instrument panel.

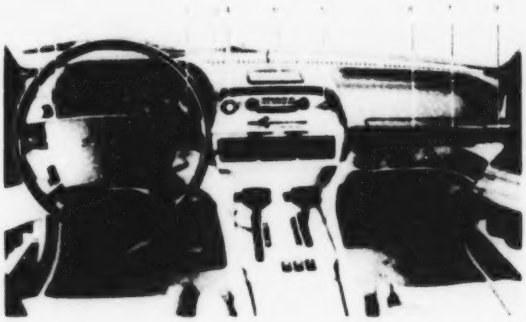


Figure 36

1. Lighting switch:
   Switch pushed in: Day intensity. Bright turn signals and brake lights.
   Switch out to first position: Headlights, parking lights, tail lights, license plate lights, instrument lights on.
   Night intensity: Reduced brightness of turn signals and brake lights.
   Switch out to second position: Foglights instead of headlights, all signal lights day intensity as with switch pushed in. Yellow warning lamp for foglights in switch knob.
   Instrument lighting by turning the knob.
2. Switch for adjustment of steering wheel and pedals to suit individual drivers.
3. Pushbutton switch for co-pilot.
4. Pushbutton switch for emergency warning lights with red indicator lamp in knob.
5. Cigarette lighter.
6. Recessed ashtray with magnetic lid.
7. Drawer-type glove compartment with integral foam padded facing.
8. Rotary latch for glove compartment has recessed button and mass compensation to prevent opening in a crash.
9. Pushbutton for hydro-pneumatically operated footrest for passenger.

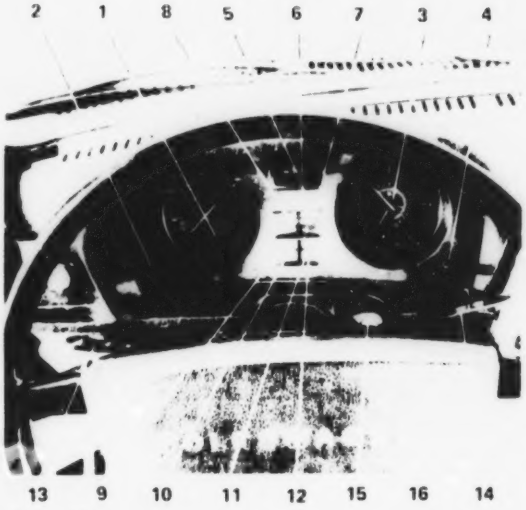


Figure 37

10. Vents for windshield defrosting and ventilation.

Instrumentation and Steering Column Controls (Figure 37)

1. Speedometer with total and trip odometer, glare-free due to curved and shrouded glass.
2. Zeroing button for trip odometer.
3. Electric clock with pre-set timer for heater, glare-free.
4. Knob to set clock and to pre-set heater.
5. Fuel gauge with reserve indicator light.
6. Turn signal indicator (green).
7. High-beam indicator (blue).
8. Vehicle spacing and warning lamp (for planned radar).
9. Bulb failure indicator (red).
10. Warning lamp for:
    Dual-circuit brakes (red)
    Brake lining wear
    Handbrake
11. Oil-pressure warning lamp (red).
12. Battery charge warning lamp (red).
13. Steering column lever, left: Radial movement upwards, right-turn signal; Radial movement down, left-turn signal.
    With ignition off: right or left parking lights.
    Lifted towards steering wheel: high or low beam and headlight flasher.
14. Steering column lever, right: windshield and headlight wiper control. Windshield and headlight washer control.
15. Ignition switch.
16. Impact pad and horn control.

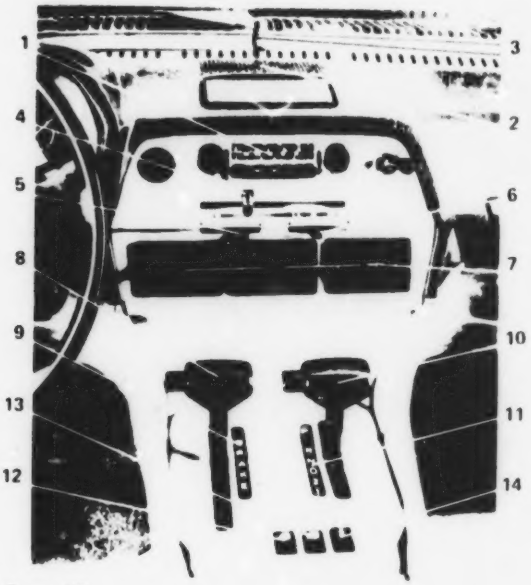


Figure 38

Central Console (Figure 38)

1. Radio.
2. Loudspeaker.
3. Lead-in for windshield antenna.
4. Heater control.
5. Ventilation control, left.
6. Ventilation control, right.
7. Fresh air outlet.
8. Handbrake.
9. Handbrake warning (BRAKE illuminated red).
10. Selector for automatic transmission.
11. Selector lever indicator (illuminated green).
12. Switch for power window, left.
13. Switch for power window, right.
14. Switch for heated rear window.

Passing Times

The required acceleration time of less than 12 seconds for 50 - 110 km/h (VDA-ESP) or for 30 - 70 mph (U. S. - ESV) was not attained (Figure 39).

The ESVW times were 15.3 seconds (50 - 110 km/h and 16.3 seconds (30 - 70 mph) accelerating with kickdown. Measurements made with a standard vehicle converted to ESVW components, dimensions and weights.

Emissions

The engine is an enlarged air-cooled VW Type 411 engine. It has a capacity of 2.0 liters and an output of 100 DIN HP at maximum speed. To keep emissions to a minimum, an electronic fuel-injection system was used.

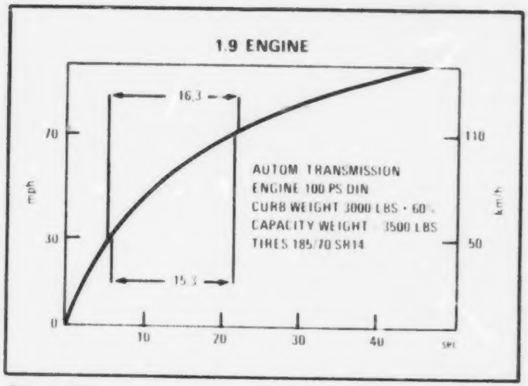


Figure 39

The vehicle has a VW automatic transmission. In a CVS cold-start test (2-bag test) the ESVW produced the following emissions:

CO	28.1 g/mi
HC	3.0 g/mi
NO_x	3.0 g/mi

The 1973 U. S. limits are (for comparison):

CO	39.0 g/mi
HC	3.4 g/mi
NO_x	3.0 g/mi

CRASHWORTHINESS

The structure and restraint system of the ESVW are designed in such a way that they fulfill the performance levels of the U. S. Specifications with two exceptions:

1. For side impacts, resultant accelerations in the chest and pelvis were higher than the 20 g of the U. S. Specifications.
2. For the rear impact, the speed of the moving barrier or car was 30 to 40 mph, a reduction from the U. S. Specifications of 30 to 50 mph.

The reasons for the two reductions are:

1. There is no biomechanical data known to VW to justify the drastic lowering of tolerable acceleration in the side impact from the tolerable 60 g in the frontal impact.
2. Benefit-cost analysis shows that the rear impact speed must be reduced considerably to obtain the same benefit-cost factor as for the additional expenditures in the front and side structures [3], [4].

The crashworthiness of the ESVW was proved experimentally in the following tests:

10 mph	frontal fixed pole or barrier impact without any damage.
20 mph	head-on car-to-car collision without any damage.
30 mph	45° frontal fixed barrier impact.
50 mph	0° - 15° frontal fixed barrier impact.
50 mph	0° frontal fixed pole impact.
75 mph	head-on car-to-car collision.
15 mph	pole side impact.
30 mph	0° and 45° car-to-car side impact.
5 mph	rear-end fixed pole or barrier impact without any damage.
40 mph	rear-end car-to-car collision.
600 mm	inverse drop test for roof structure.

Structure

The supporting and energy-absorbing structure of the ESVW consists of a framework integrated into the unit body (Figure 40).

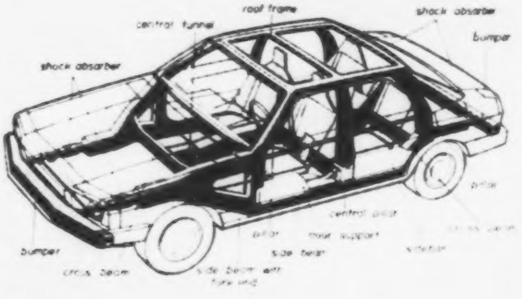


Figure 40

In a frontal impact, the forces are transmitted through the bumper, the shock absorbers and the fork to the A-pillars. From there, the forces travel to the roof members, the side bars and the sill panels and are brought into equilibrium by the inertia forces of the rear end.

In a side impact, the forces are transmitted through the strong A, B and C-pillars and the cross members to the pillars on the opposite side of the passenger compartment. The cross members between the B and C-pillars are integrated in the seats.

This concept gives the maximum possible lateral stiffness with medium weight increase, but requires fixed seat backrests for the front seats and thus necessitates an adjustable steering wheel and pedals.

To control the concentrated load penetration in a pole impact or angular vehicle impact at front and side, the front and sides are provided with a so-called "hard edge" by using particularly strong bumpers and side bars. The box section for the bumpers is about 130 x 160 mm (height x width) and for the side bars it is 140 x 140 mm. Experiments with heat-treated aluminum malleable alloys did not bring the desired results because of the low breaking elongation under impact. High quality micro-alloyed hot strip steel with a yield strength of 46 kp/mm^2 and a breaking elongation of about 25% was used. The metal thickness selected for the bumpers was 3.5 mm and that of the side bars was 2.0 mm. Bumpers and side bars are so dimensioned that a pole impact causes localized denting and bending but the remaining cross section can still transmit the necessary forces. At the load penetration locations the hollow sections are reinforced by bulkheads. The attachment of the side bars to the pillars is designed so that forces are transmitted in the transverse direction as well as in the longitudinal direction.

High quality steel is also used in the forks, the pillars, the cross members and the roof frame in addition to the bumpers and side bars. The general principle appears to be that the use of high quality steel in the rigid passenger compartment and the "hard edge" is advantageous but not in the energy-absorbing zones at front and rear.

One special feature of the ESVW is that a front or rear engine can be installed without modification of the outer panels and without major modification of the running gear and the general structure (Figure 41).

The theoretical deformation distances without penetration of the passenger compartment are more than 0.82 m at the front, 0.2 m at the side and 0.6 m at the rear (front engine) or 0.35 m (rear engine).

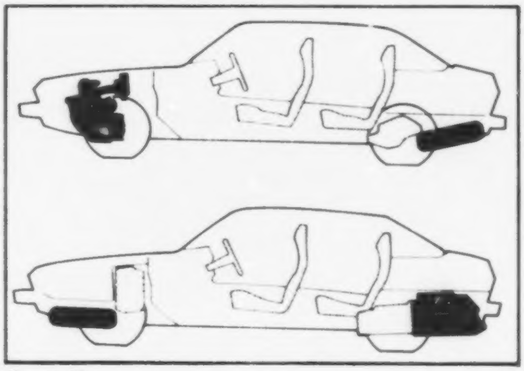


Figure 41

Occupant Crash Protection And Vehicle Aggressivity

In the development of the ESVW the two aspects of crashworthiness — occupant crash protection and vehicle aggressivity — were taken into account to the same extent. This was based on the belief that an over-all increase in passive safety cannot be brought about by exceptionally heavy, large and rigid vehicles. Instead, because of the considerable number of collisions between light and heavy vehicles, any over-all increase in safety must come from vehicles which in mass, rigidity and design have the least aggressive behavior in relation to their potential collision partners.

This principle requires that the large vehicles have a "soft nose." Based on the U. S. Specifications, Volkswagenwerk has made a suggestion for the design of impact-velocity sensitive front structures on vehicles of different weights using hydraulic shock absorbers [3] [4].

The results are shown in Figure 42 as the increase in hydraulic stroke d' of the absorber with the vehicle mass in relationship to the vehicle mass.

Figure 43 shows the decelerations of cars of different weights in a head-on fixed barrier impact versus the impact velocity.

This representation is so far ideal in that it is based merely on energy balance and assumes a constant force deformation curve. This type of behavior is achieved better with "Orifice absorbers" than with "Valve absorbers" [5].

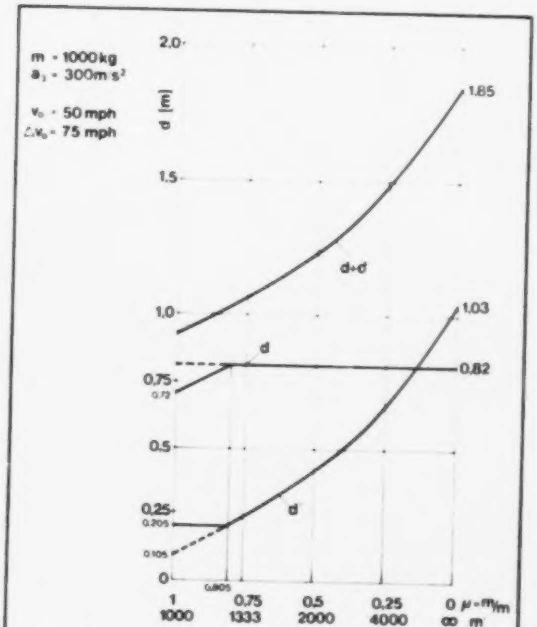


Figure 42

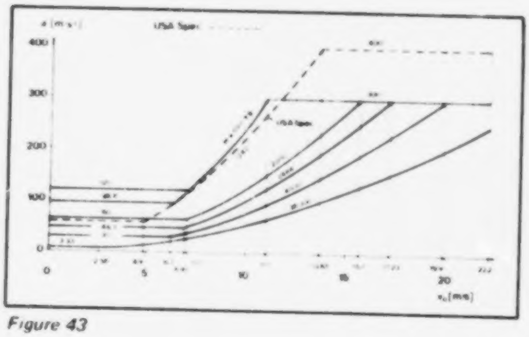


Figure 43

The dynamic process in a 50-mph fixed barrier impact with a 1,400 kg car, equipped with orifice absorbers or valve absorbers with the ideal stroke of 0.25 m as in Figure 42 is shown in Figures 44 and 45.

h represents the hydraulic stroke,

s the plastic deformation and

v the speed of the vehicle.

The dynamic process in a frontal collision between a 1,400 kg car and a 2,460 kg car, equipped with orifice absorbers with the ideal stroke length as in Figure 43 is shown in Figure 46. As the large vehicle absorbs a considerable part of the energy, the tolerable deformation of 0.82 m for the small vehicle is not exceeded. If one varies the mass of the larger

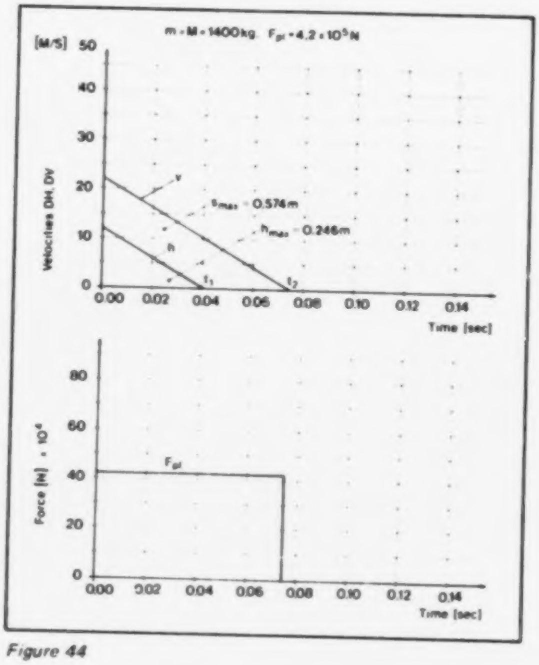


Figure 44

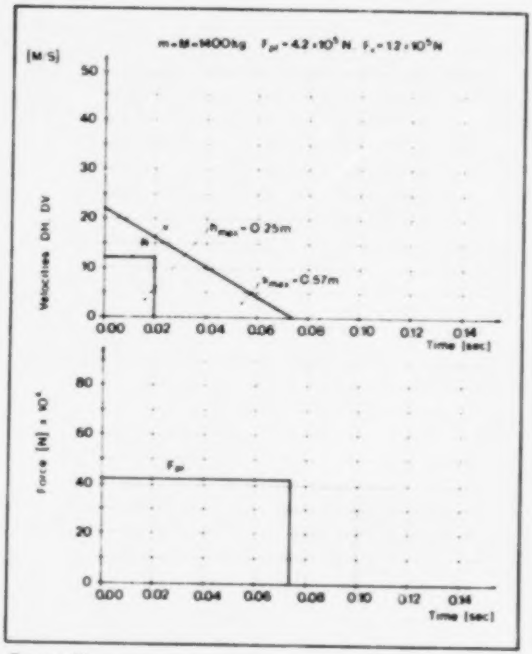


Figure 45

vehicle, the permissible relative speed at which the total deformation of the small car is just 0.82 m drops with the increasing mass (Figure 47). On the whole, the objective of non-aggressivity of vehicles in a wide mass range is achieved by the use of hydraulic absorbers.

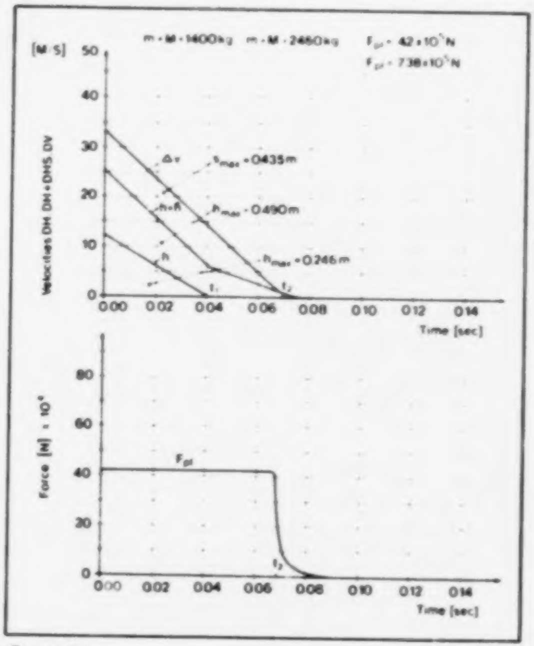


Figure 46

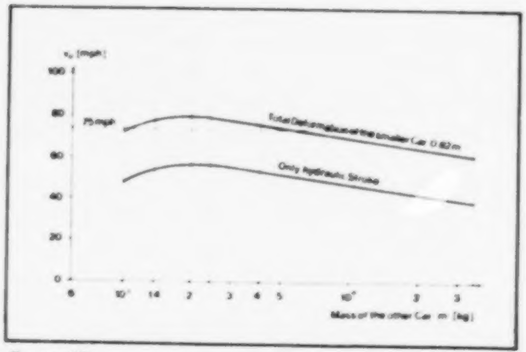


Figure 47

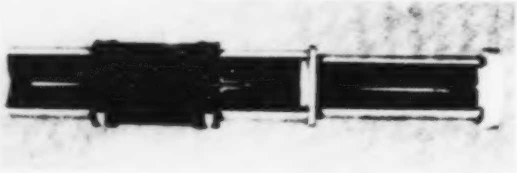


Figure 48

Deviating from the most favorable theoretical design orifice absorbers with a 0.25 m stroke a slight modification was made on the ESVW for installation reasons and a combination of orifice and valve absorbers (Figure 48) with a stroke of 0.205 m was used. In the 10-mph impact there is a reversible stroke of 0.1 m. At the rear, orifice absorbers with 50-mm strokes were used and these absorbed an impact at 5 mph against a fixed barrier reversibly.

Test Results

The crash tests were carried out with ESVW prototypes in the form of test frames (Figures 49 and 50) with rear engines. For the vehicle-vehicle collisions, a converted VW K 70 which has a front engine was used. The test frames simulate the structure of the ESVW and are almost identical to the prototype with respect to the geometric dimensions and material. They consist merely of straight channel and sheet metal parts and are therefore easier and quicker to manufacture and also to modify, where necessary, than prototypes made of more complex pressed parts. For some of the tests, the frames were only fitted with such parts as were necessary for the simulation of energy absorption and strength. For the rear-end collision, for example, the front hydro-absorbers, bumpers, wheel housing, fenders, steering wheel and tank were not fitted. In addition to four test dummies, further masses were added to bring the

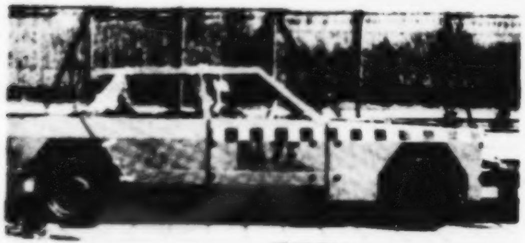


Figure 49

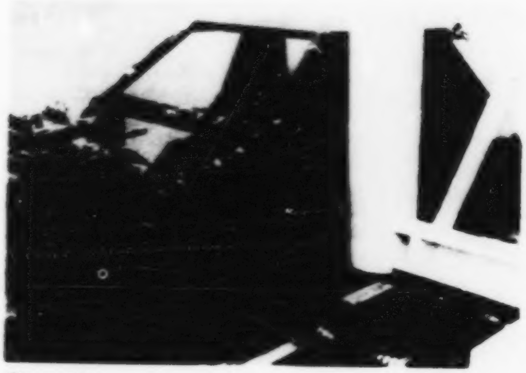


Figure 50

total weight up to 1,400 kg. The total weight was deliberately kept below 1,700 kg to compensate for the disadvantages of the frames as opposed to the ESVW in respect of strength and energy absorption.

To test our point of view concerning the reduction of heavy vehicle aggressivity, all vehicle-vehicle collisions were carried out with a vehicle weighing 2,500 kg. This vehicle was simulated by a mobile barrier, the bumpers of which are equipped with orifice hydro-absorbers with a 0.5 m stroke (Figures 42 and 56). In the vehicle-vehicle collisions described here, this stroke is fully utilized in every case as can also be shown theoretically [5] (Figure 46).

The following report deals only with the main tests and in each case shows the vehicle before the test, the vehicle after the test and the deceleration/acceleration diagrams for each vehicle.

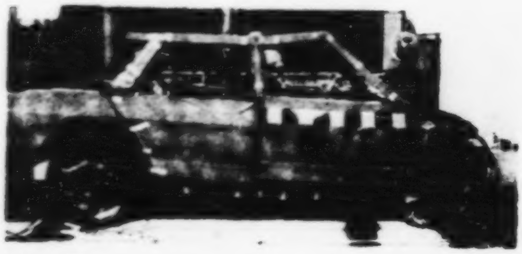


Figure 51

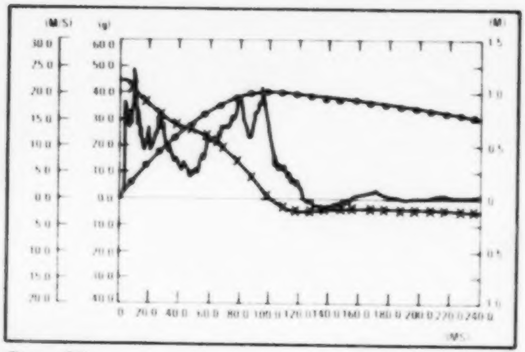


Figure 52

50-mph Frontal Fixed Barrier Impact (Figures 49, 51, 52)

The mean deceleration figure of 24 g is below the figure of 30 g being aimed for, the deformation of 1.03 m is above the design target of 0.82 m.

50-mph Frontal Fixed Pole Impact (Figures 53, 54, 55)

Figure 53 shows the frame before the crash and Figure 54 after the crash. The mean deceleration figure is 20 g (Figure 55). The dynamic deformation of 1.2 m can also be seen. No intrusion into the passenger compartment or tank deformation took place. In this version, the test frame just about reached the design targets of 30 g and 0.82 m in the frontal fixed-barrier crash.

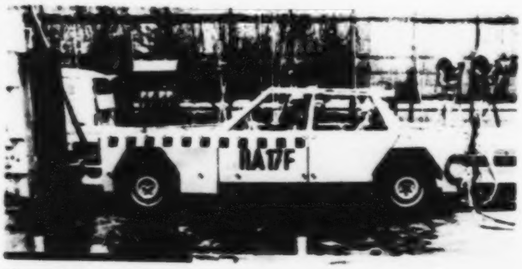


Figure 53

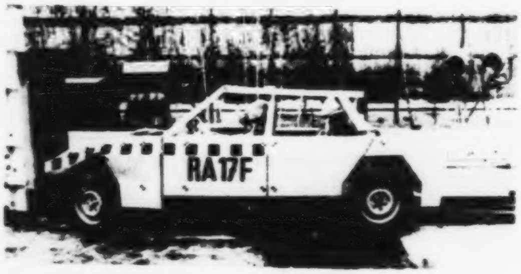


Figure 54

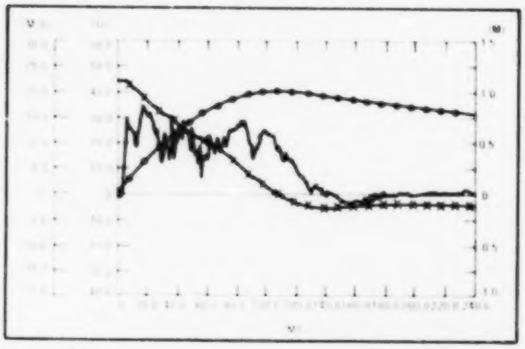


Figure 55

75-mph Head-on Car-to-car Collision (Figures 56, 57, 58)

The deceleration time curve is almost identical to that of the 50-mph barrier impact with a K 70 converted in the same way. The deformation was slightly less. The test result confirms that the occupants of lighter vehicles can survive collisions with heavier vehicles if the heavier vehicle deforms adequately, that is, if it is not more aggressive than is permissible.

40-mph Rear End Car-to-car Collision (Figures 59, 60, 61)

The deformation of the rear end was 200 mm with an average acceleration level of 20 g. No intrusion into the passenger compartment took place.

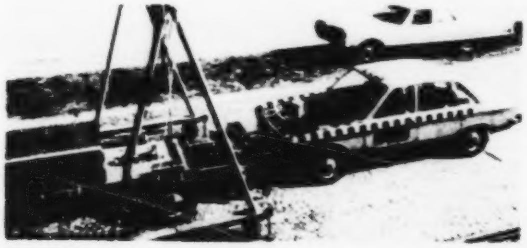


Figure 56

Figure 57

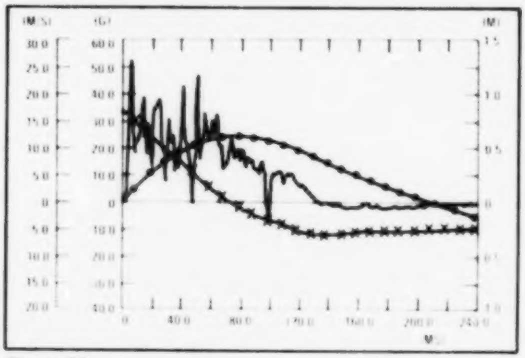


Figure 58

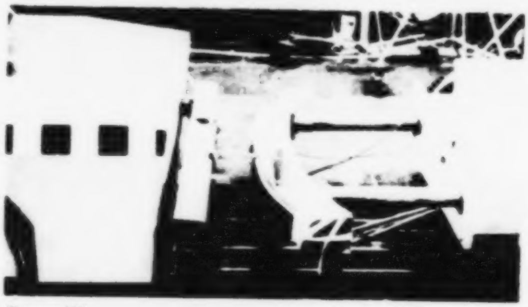


Figure 59

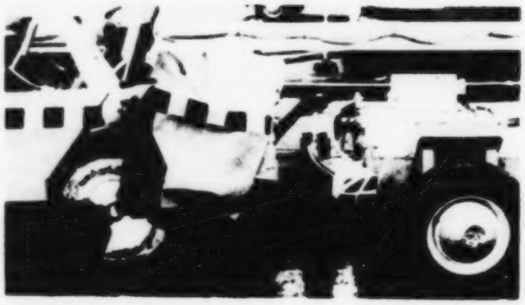


Figure 60

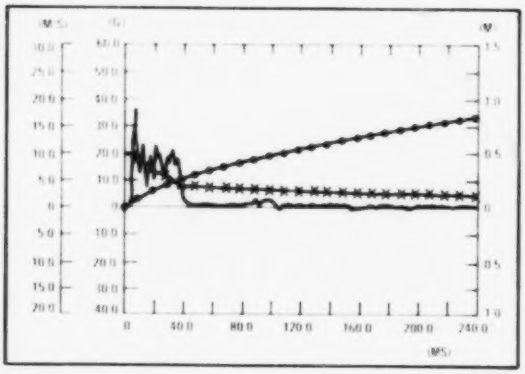


Figure 61

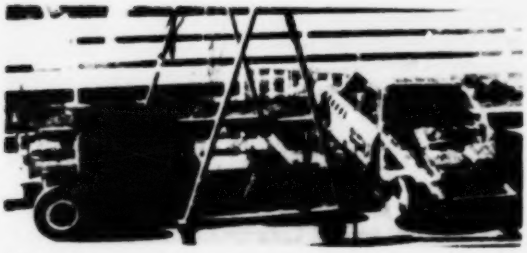


Figure 62

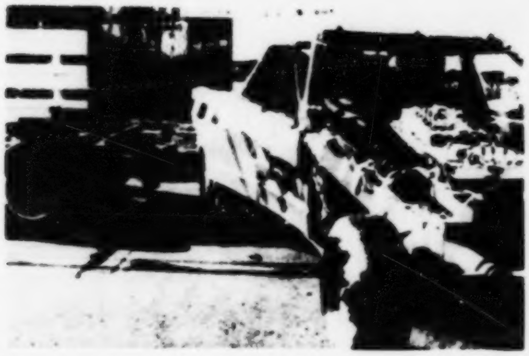


Figure 63

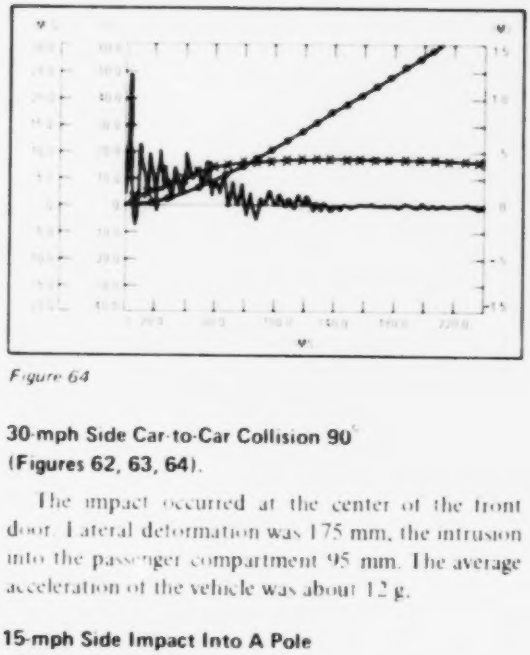


Figure 64

30-mph Side Car-to-Car Collision 90°
(Figures 62, 63, 64).

The impact occurred at the center of the front door. Lateral deformation was 175 mm, the intrusion into the passenger compartment 95 mm. The average acceleration of the vehicle was about 12 g.

15-mph Side Impact Into A Pole
(Figures 50, 65, 66)

The vehicle struck the pole at the center of the front door. In this case the deformation was 260 mm and the intrusion 110 mm. The mean g-level again was about 12 g.

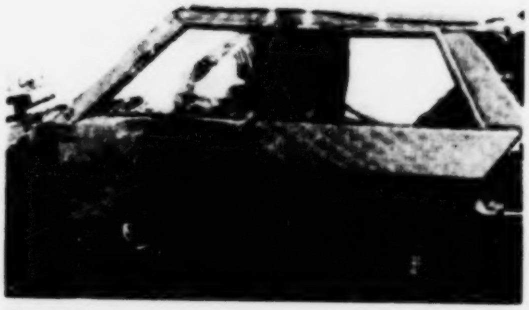


Figure 65

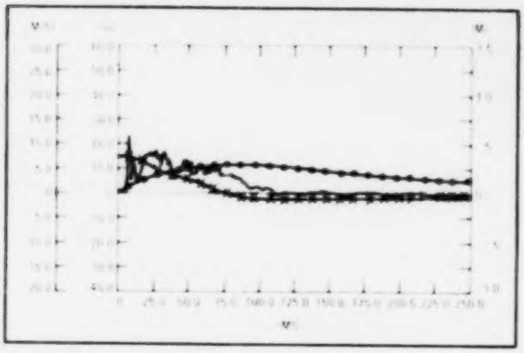


Figure 66

Other tests described in the U. S. - ESV Specifications, such as:

50 mph 15° frontal fixed barrier impact
30 mph 45° frontal fixed barrier impact
30 mph 45° car-to-car side collision

were carried out with test frames or converted K-70 vehicles.

In all cases the specifications were met.

RESTRAINT SYSTEM

In the development of the safety vehicle, particular concentration was devoted to the design of the restraint elements for the occupants. The criteria for the restraint system were:

At the head-on collision speeds required by the specifications, the occupant of the vehicle must be protected from injury by keeping mechanical strains to a minimum (accelerations and forces), and forward movement must be so small that the passenger does not impact the vehicle components in front of him. When the system becomes operative it must not result in injury to the occupant. The system must be passive, requiring no action by the occupant.

Any loss of comfort or convenience in driving caused by the system must be kept to the unavoidable minimum.

Theoretical Considerations [14]

Figure 67 shows the velocity versus time histories of the passenger compartment (Vv) and of the occupant (Vp) from the moment of impact until the vehicle is stationary. The shaded area is equivalent to the displacement of the occupant relative to the vehicle. The size of this area is dependent on the extent of the average occupant deceleration and the average vehicle deceleration, the time delay (t_1) before occupant deceleration begins after the vehicle deceleration has started.

Figure 68 shows the forward displacements for two different reaction times applied to the impact velocity and for various average occupant decelerations. Derived from this in Figure 69, we see the relationship of the displacement to the time delay for a passenger compartment deceleration of 30 g and an average occupant deceleration of 40 g, following a 50-mph frontal collision.

With a conventional belt, the time delay t_1, resulting from elasticity in the belt system, in the clothing and in the human body, as well as belt slackness, is in the region of 30 milliseconds. With airbags of present design the time delay is about 40 milliseconds.

The design goals are therefore the following:

- pretensioning of a belt at the start of the crash to reduce the time delay.
- force limitation in the belts, providing increased absorption of distortion energy and thus a controlled retardation of the occupants after the start of the crash.
- reduction of the local body-to-belt loads.

System Description

The restraint system used in the ESVW consists of a diagonal shoulder belt and a knee belt. The shoulder belt is of oval cross-section and is foam filled. This design was selected to reduce the local surface pressures such as occur under the hard edges of flat belts, to reduce the risk of injury in the neck area, to provide a certain rigidity for passive fitting and to prevent twisting of the belt. From the force limiter (a plastically deformable torsion bar at shoulder level) on the B-pillar, the belt runs through a chrome ring inside the center tunnel and from there forward to the lower portion of the instrument panel. At this point, the belt beomes the knee belt and is three times as wide as a conventional belt. This wider flat

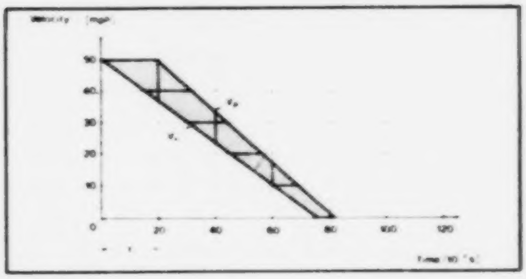


Figure 67

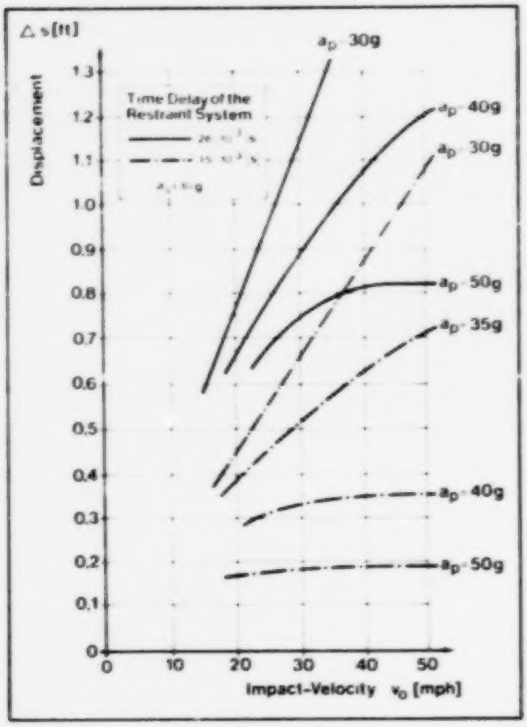


Figure 68

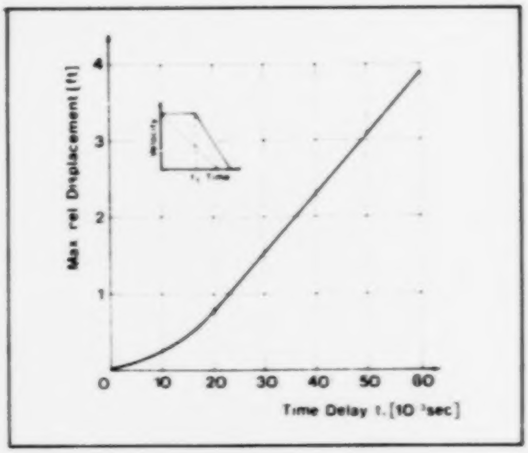


Figure 69

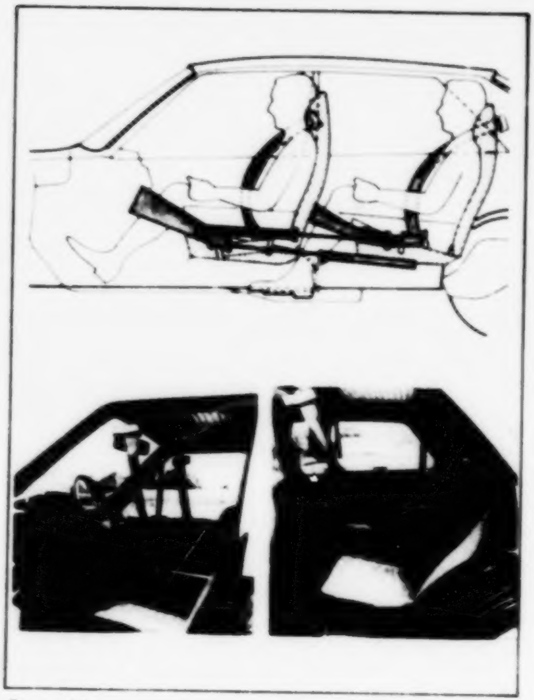


Figure 70

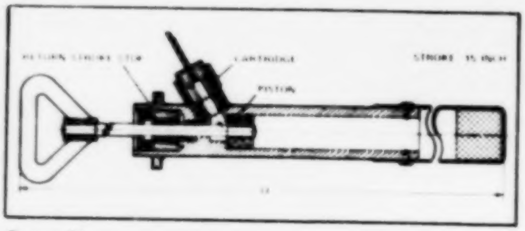


Figure 71

belt is concealed under the instrument panel and then continues to a second force limiter inside the door (Figure 70). To facilitate entering or leaving the vehicle, the shoulder level mounting of the belt is moved along a guide rail above the door opening by vacuum action. On closing the door and starting the engine, the partial vacuum in the intake manifold is used to move the upper end of the shoulder belt to its "active position" at the B-pillar. When the door is opened a venting valve opens, releasing the vacuum and the belt is pulled back along the rail toward the A-pillar by spring action so that it does not obstruct entry or exit. The part of the knee belt that runs inside the door releases when the door opens and retracts as the door is closed.

In the "active position" the belt provides a 95% man with sufficient slack for him to reach all operating controls. Because the seat does not adjust fore and aft, smaller persons will have a greater amount of slack and can reach farther. Because of this, the shoulder belt does not hinder driving operations. In the normal state, the knee belt is fitted behind a breakaway fabric in the lower part of the instrument panel and therefore does not restrict the movements of the occupants. Belt preloading takes place at the instant a frontal impact of over 15 mph occurs. This eliminates the injurious effect of a slack belt on impact and causes the occupant to participate in the vehicle deceleration from the start of the crash. The entire belt system is drawn tightly to the chest and knees.

On a frontal impact of over 15 mph, a pressure switch in the hydraulic bumper triggers a cartridge inside the belt preloader, retracting the piston to deploy the knee belt and preload the entire system. As the occupant loads the belt during deceleration, a return stroke stop prevents the piston from returning to the slack position (Figure 71).

An instant-disconnect button on the upper force limiter of the belt permits the occupant to release himself after the impact. Since the occupant protection provided by the belt system is quite sufficient without preloading in low-speed impacts, tension is only applied at speeds corresponding to a 15-mph barrier crash. This is accomplished by regulating the activating pressure of the switch.

With this belt system, the time delay between the start of impact and application of 50% maximum belt force is about 12 milliseconds. At speeds below the 15-mph trigger threshold, the occupant is held by the diagonal belt with its force limiter and by an energy-absorbing cushion at knee impact level.

Sled Test Results

Sled tests were used to prove the effectiveness of the restraint system. The test sleds were braked with virtually square wave deceleration to simulate the vehicle retardation, the g level being adjustable in accordance with the collision speed.

In actual practice, the values used were as in Figure 72. The test results with 50% male dummies are shown in Figures 73 - 77 for the resultant head acceleration, the severity-index, the resultant chest acceleration, the resultant pelvis acceleration and the femur loads. The shaded area in the Figures 73 - 77 shows the results of frontal impacts at speeds below 15 mph. In these cases no preloading takes place.

The reproducibility of the sled tests is ensured even for 50-mph impact speeds, as shown in Figures 78 and 79, of the acceleration to time curves for the head and chest.

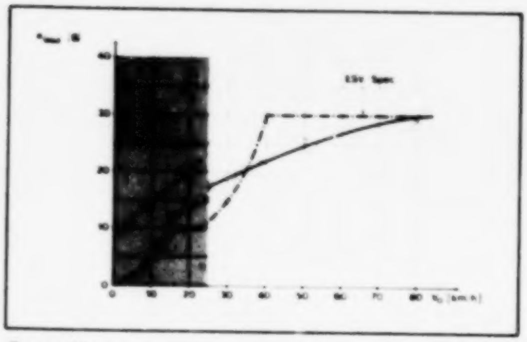


Figure 72

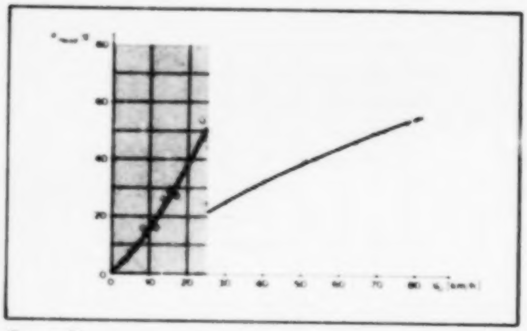


Figure 73

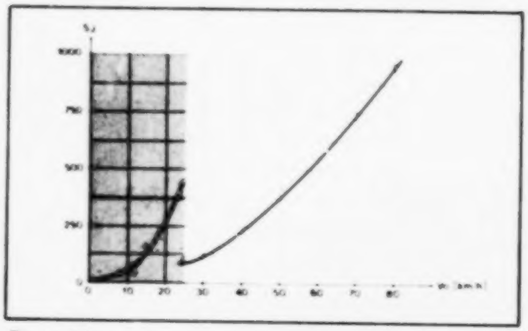


Figure 74

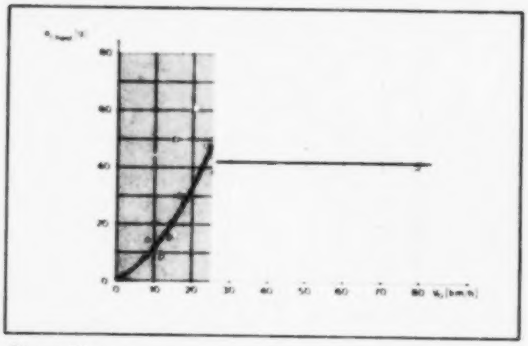


Figure 75

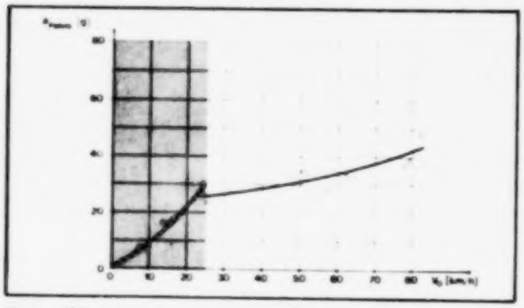


Figure 76

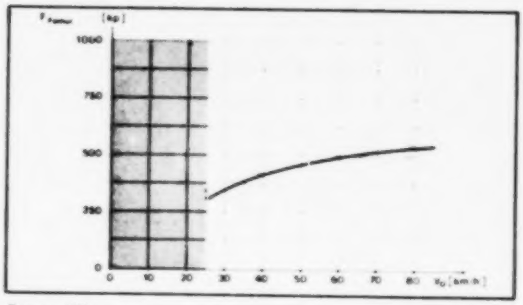


Figure 77

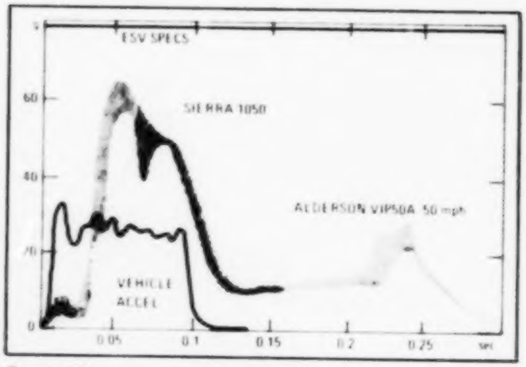


Figure 78

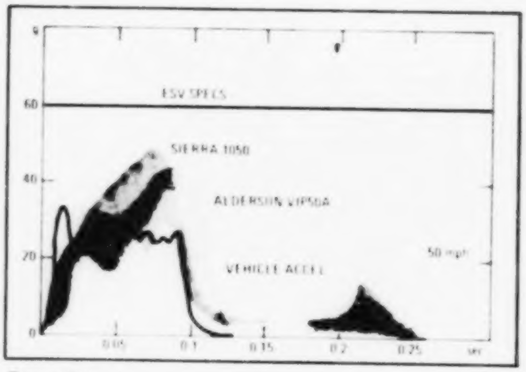
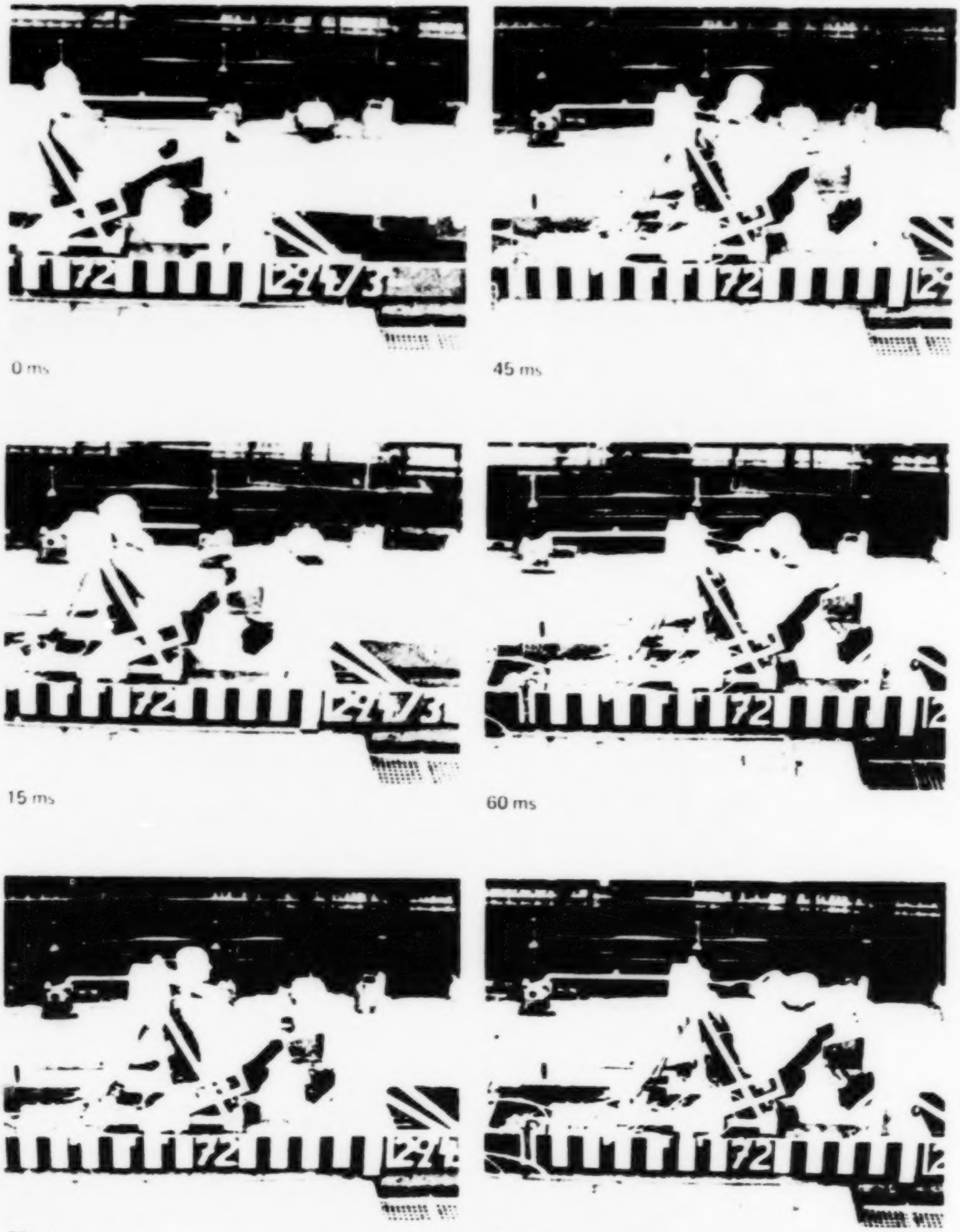


Figure 79

The Movements Of The Test Dummy During A 50-mph Sled Test Is Seen In Figure 80a.



0 ms

15 ms

30 ms

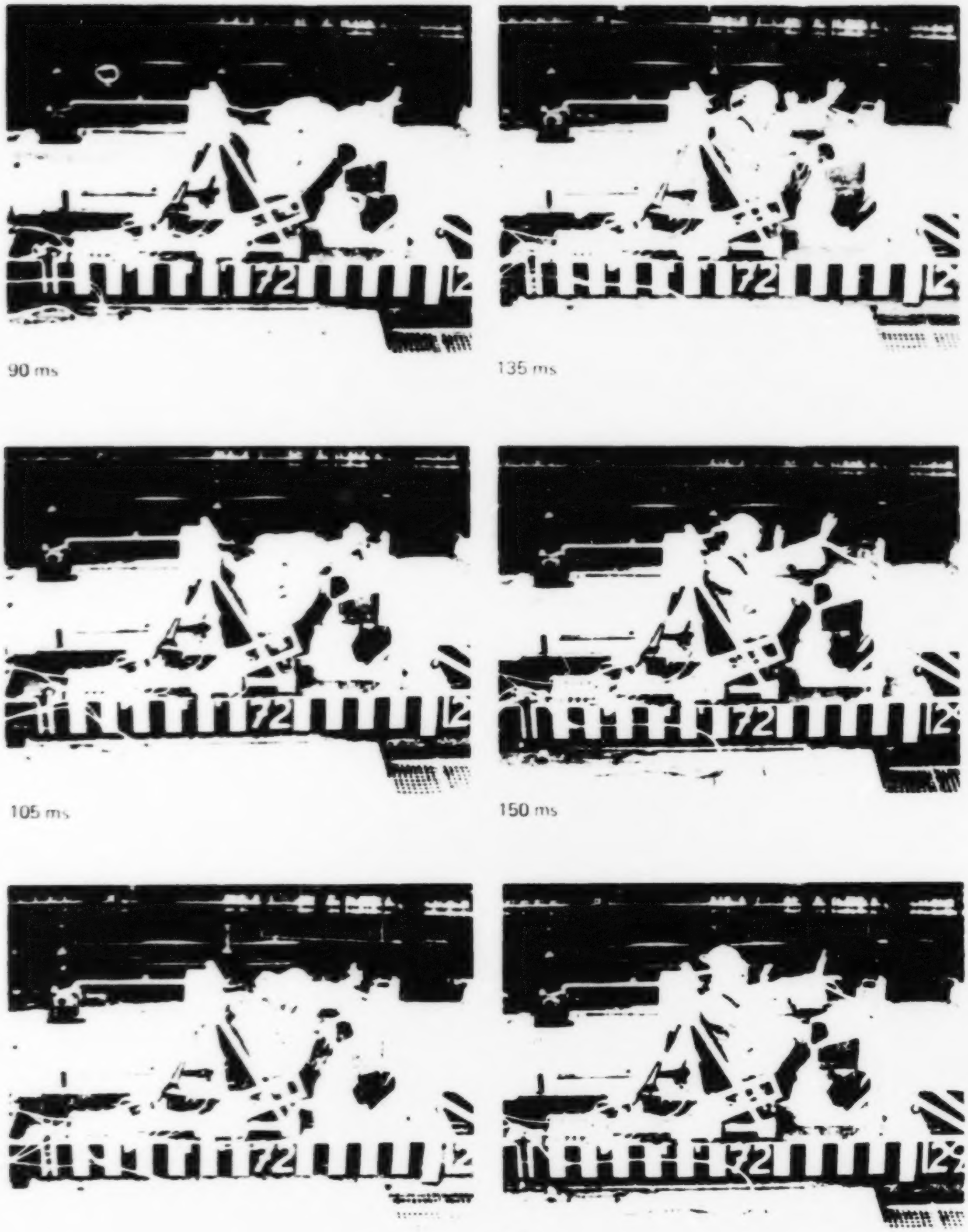
45 ms

60 ms

75 ms

Figure 80a

The Movements Of The Test Dummy During A 50-mph Sled Test Is Seen In Figure 80b.


72
12
72
12
72
12
72
12
72
12
72
12

90 ms

135 ms

105 ms

150 ms

120 ms

165 ms

Figure 80b

Reproduced from best available copy.

Vehicle Test Results

The ESV frame tests (as described above) conducted as evidence of crashworthiness were combined with occupant restraint system tests. In this way it was possible to prove that requirements in respect of occupant protection in head-on, lateral and rear-end collisions were met.

Extreme strains on the structure and restraint system occur at 50 mph in head-on collisions against the fixed barrier and against a 14-inch pole.

The following Tables 5 and 6 contain the result of both tests:

Table 5. ESV Frontal Barrier Crash at 50 mph.

Passenger	Acceleration head, max (g)	Requirement USA VDA	Acceleration chest, max (g)	Requirement USA VDA	Acceleration pelvis, max (g)	Requirement USA VDA
50% male front left	78	fulfilled	47	fulfilled	55	fulfilled
5% female front right	79	fulfilled	46	fulfilled	—	—
5% female rear left	71	fulfilled	37	fulfilled	—	—
50% male rear left	44	fulfilled	47	fulfilled	40	fulfilled

Table 6. ESV Frontal Pole Crash at 50 mph.

Passenger	Acceleration head, max (g)	Requirement USA VDA	Acceleration chest, max (g)	Requirement USA VDA	Acceleration pelvis, max (g)	Requirement USA VDA
50% male front left	70	fulfilled	35	fulfilled	42	fulfilled
5% female front right	79	fulfilled	40	fulfilled	49	fulfilled
5% female rear left	70	fulfilled	36	fulfilled	51	fulfilled
50% male rear right	40	fulfilled	35	fulfilled	41	fulfilled

Table 7. The Occupant Forces on a 90° Car-to-car Side Collision at V = 32.8 mph. (52.5 km/h).

Occupant 50% male dummy	Acceleration head, max (g)	Requirement USA VDA	Acceleration chest, max (g)	Requirement USA VDA	Acceleration pelvis, max (g)	Requirement USA VDA
front left	no readings available					
front right	35	fulfilled	24	fulfilled	32	fulfilled
rear left	24	fulfilled	21	fulfilled	29	fulfilled
rear right	40	fulfilled	28	fulfilled	24	fulfilled

Table 8. The Occupant Forces on a Car-to-car Rear-end Collision at $V_{rel.}$ = 43 mph (69.2 km/h).

Occupant 50% male dummy	Acceleration Head max (g)	Requirement USA VDA	Acceleration Chest max (g)	Requirement USA VDA	Acceleration Pelvis max (g)	Requirement USA VDA
front left	32	fulfilled	20	fulfilled	27	fulfilled*
front right	33	fulfilled	30	fulfilled*	24	fulfilled*
rear left	39	fulfilled	17	fulfilled	43	fulfilled*
rear right	29	fulfilled	19	fulfilled	22	fulfilled*

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*The U.S.-requirement of 20g (see page 126) chest and pelvis was not fulfilled (see Tables 7 and 8). These limits were raised by VW to 60g.

BENEFIT-COST CONSIDERATIONS

After the work on the ESVW had shown in the first stage that the extreme test requirements could be met technically, effort was concentrated in the second stage on the question of economic efficiency. That is, we investigated the question of the benefits and costs on an ESV built as a series production vehicle and which hypothetically dominated the market. In the benefit-cost analysis, various components and complete vehicles were investigated. Some of the results obtained are reported upon [6].

In the following detail, the benefit of the safety measures is considered only economically. It is computed by determining the monetary value of fatal and injury casualties that would be eliminated by the ESV. In [7] the economic benefits of these was computed with these values $43,000 per fatality, $2,200 per injured person.

The total figures used are from United States statistics for passenger-cars occupant casualties for the year 1969 [5]:

Deaths USA 1969: 41,630
Injured USA 1969: 1,731,000

From this, for the USA the total economic damage for 1969 amounts to 5.6×10^9 $. The costs quoted in this section are estimates and should be regarded as being the lower limits.

RESTRAINT SYSTEMS

Consideration has been given to fitting four seats with:

3-point belts,
3-point belts with force limiters,
3-point belts with force limiters and preloading,
air bags.

With the 3-point belts, differentation is also made between active, active-passive (interlock system) and passive versions see Table 9.

The effective efficiency of a restraint system is obtained from the product of the theoretical efficiency and reliability or percentage of users see Table 9. The assumption of a theoretical effectiveness of 45% for 3-point belts and of 30% for air bags is used on data in various literature [3] [8] [9] [10] [11] [12] and our own investigations.

Total annual benefits are obtained as the product of effective efficiency and total damage. The benefits per car life are then obtained by multiplying by the average vehicle life (10 years) and dividing by the total vehicle population (90 million).

Table 9 Benefits and Costs of Restraint Systems

Restraint System		Reduction of Deaths and Injured %	Estimated Use, Reliability %	Effectiveness %	Benefits: Absolutely 10⁶ $	Benefits: per Car LT (Life Time) $	Costs: Retail per Car $	Costs: Maintenance p. C. LT. $	Costs: Structure per Car $	Costs: Total p. C. LT. $	Benefit/Cost-Ratio
Active	4 3-Point Belts	45	10	4.5	252	28	80	27		107	0.26
	3-Point Belts + Force Limiter	50	10	5.0	280	31	100	34		134	0.23
Interlock	4 3-Point Belts	45	70	31.5	1,764	196	93	31		124	1.6
	3-Point Belts + Force Limiter	50	70	35.0	1,960	218	113	38		151	1.4
	3-Point Belts + Preloading	55	70	38.5	2,156	240	191	64	38	293	0.82
	3-Points Eelts + Enforced Use	55	80	44.0	2,464	274	191	64	38	293	0.94
Passive	4 3-Point Belts	45	90	40.5	2,268	252	124	42		166	1.5
	3-Point Belts + Force Limiter	50	90	45.0	2,520	280	144	48		192	1.5
	3-Point Belts + Preloading	55	90	49.5	2,772	308	222	74	38	334	0.92
	4 Air Bags	30	90	27.0	1,512	168	320	107	46	473	0.36

The costs are composed of the procurement costs of the restraint system, the costs of various installation work (including seat reinforcement and knee bars) for belts with and without preloading, and for air bags, together with the maintenance costs. The maintenance costs are presumed to be annually 5% of the procurement costs and accounted for at an interest rate of 8% on the initial value over the vehicle life of 10 years – see Table 9.

Benefits are shown graphically in Figure 81 as functions of the costs. From this it will be seen that:

- Active 3-point belts have a poor benefit-cost ratio of 0.26 due to the low percentage of users (10%).
- The greatest benefit-cost ratio of 1.6 is seen on the 3-point belt with the interlock system.
- The maximum advantage where advantage = benefits minus costs is obtained from a passive belt system.
- Air bags are superior to active 3-point belts. They are inferior to 3-point belts with interlock systems in the benefit-to-cost ratio.

CARS WITH IMPROVED STRUCTURES

In this section, the vehicles considered have been improved in comparison with present-day vehicles in their structures with regard to various combinations of tests. It is assumed that the restraint system is a passive 3-point belt with a force limiter and preloading or air bags.

In the third stage of the MVSS 208 which is to become effective in the United States August 15, 1975, these conditions are laid down for testing the functional safety of the passive restraint system and of the vehicle structure:

The test conditions are 30/20/30, i.e., 30 mph head-on fixed barrier impact, (0° – 30°), 20 mph side impact by a moving barrier of 4,000 pounds; 30 mph rollover.

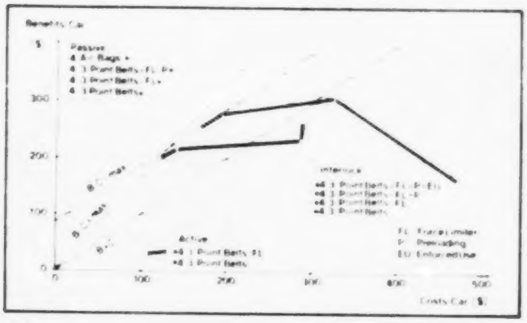

Figure 81

In the considerations which follow, it is necessary to take into account the rear-end structure. Because of this, another, more or less equivalent, combination of accident types has been selected: head-on fixed barrier impact; side impact by identical car; rear end impact by identical car.

In the NHTSA program, the MVSS 208 is regarded as the first of three phases. The speeds in the various test combinations are, including the ESV:
30/20/30 mph
45/30/45 mph
50/30/50 mph (ESV test combination)
60/40/50 mph

Under present safety standards, passenger cars such as the VW K 70 (unladen weight 1,050 kg = 2,315 pounds) must be tested at the following speeds: 30/15/30 mph.

Taking the K 70 and ESVW for which the weights and performances are known as points of reference, it is possible to calculate the weights and costs needed for the various test combinations under the following assumptions:

1. Consideration has only been given to the additional weights in the front, side and rear necessary to convert a vehicle of present-day design for the increased test speeds.
2. The additional costs are proportioned to the weight increases and the weight increases to the additional energy absorption at the front, side and rear.
3. The differences in weight between the ESVW and the VW K 70
   105 kg = 232 pounds at the front
   115 kg = 254 pounds in the sides and roof
   58 kg = 128 pounds in the rear
   are just those weights which are necessary to convert the K 70 to meet the U.S.-ESV Specifications.
4. Selling costs of $2.50 per kg or $1.14 per pound of additional weight have been added.
5. The payload is assumed to be 350 kg = 440 pounds.
6. The absolute additional expenditure will be more or less equal on vehicles in the 1,000 to 2,000 kg range [11].

From the comparison of the K 70 and the ESVW it may be calculated that the front and rear structural costs related to energy are approximately the same:

Front: $9/10^4 Nm
Rear: $10/10^4 Nm

The equivalent figure for the side structure is ten times as much: $100/10⁴ Nm.

This shows how expensive energy absorption in the side structure is, because the energy is absorbed, not by compression and buckling, but by bending. Now that the costs related to energy absorption have been determined, it is possible to establish the additional costs and weights for each test combination by an iterative convergent procedure:

	Additional weight kg	lb	Additional procurement cost $	Additional maintenance cost $
30/20/30 mph (MVSS 208)	27	60	61	127
45/30/45 mph	228	503	570	1073
50/30/50 mph (ESV)	278	613	698	1308
60/40/60 mph	627	1383	1566	2950

In the table, the additional maintenance costs (taxation, servicing, fuel) have also been entered proportionally to the additional weights and accounted for from the date of procurement (vehicle life 10 years, 8% interest). This thus reveals that on the ESVW about $2,000 extra expense is attributable to structural improvements alone.

The additional costs of active safety are estimated to amount to some $1,200 (anti-skid, co-pilot). This total has not been broken down into greater detail because they have not been taken into account in the benefit-to-cost analysis. To include these costs would lower the benefit-to-cost factor.

By the costs entailed in knee bars, reinforcements and interior padding [4] we then arrive at the procurement costs (structure, restraint system) and maintenance costs per vehicle as shown in Table 10. The increased burden is about $2,400.

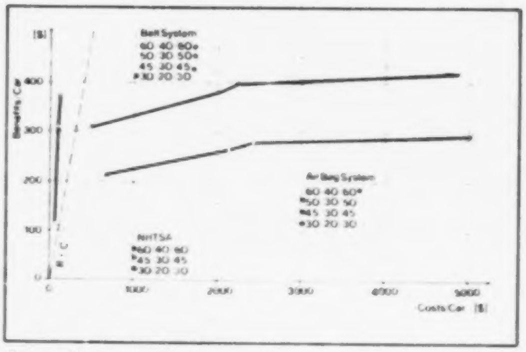

Figure 82

The effective efficiency of the passive restraint system has been estimated along with the improved structure and provides the benefits also quoted in Table 10. The ratio of the benefit-to-cost outlay is graphed in Figure 82 for the individual standards. On increasing the frontal impact speed above 30 mph it becomes clear that a small gain in safety is accompanied by severely progressive costs and that the benefit-to-cost ratio falls sharply. Furthermore, the superiority of a passive belt system over the air bag becomes obvious.

Figure 82 also includes an NHTSA estimate from 1971. In our opinion this estimate is based on unrealistic costing.

Table 10 Benefits and Costs of Crashworthiness Standards

Crashworthiness Standard	Restraint System	Effectiveness	Benefits		Add. Costs per Car		Add. Operating Cost per Car LT	Add. Cost per Car LT Total	Benefit/Cost-Ratio
			Absolutely	per Car LT	Structure	Restr. System			
		%	10⁶ $	$	$	$	$	$	
30/20/30 (MVSS 208)	Advanced Belt System	50	2.800	311	119	222	201	542	0.57
45/30/45		60	3.360	373	640	222	1147	2009	0.19
50/30/50 (ESVW)		63	3.528	392	774	222	1382	2378	0.16
60/40/60		67	3.752	417	1664	222	3024	4910	0.085
30/20/30 (MVSS 208)	Advanced Air Bag System	35	1.960	218	127	320	234	681	0.32
45/30/45		42	2.352	261	648	320	1179	2147	0.12
50/30/50 (ESVW)		44	2.464	274	782	320	1415	2517	0.11
60/40/60		47	2.632	292	1672	320	3057	5049	0.058
30/20/30 NHTSA Apr 71		20	1.120	124	75				1.7
45/30/45		50	2.800	311	100				3.1
60/40/60		60	3.360	373	125				3.0

PROPOSAL FOR FUTURE ESV SPECIFICATIONS

With first-generation test results and the other now available from the ESV program, it is possible to formulate modified specifications on the basis of what has been learned.

Future specifications ought to be consistent with each other. That means they must be intermatched with regard to their safety benefits. One method of safety benefit comparison is offered by a benefit-cost economic analysis. Such an analysis is essential particularly to those high-cost safety components. The results of a benefit-cost analysis involving only crashworthiness on the ESVW may be summarized in the following way:

1. The ESVW benefit-to-cost factor of 0.11 is well below one due to the high speed requirements and to the fact that the cost increase is more than squared with the speed.
2. To increase the energy absorption and rigidity in the side structure requires much higher costs than in the front and rear structures.
3. The safety benefit (reduction of fatalities and injuries) resulting from reinforcement of the rear structure is slight compared to the front and side structures. In keeping with the concept of "consistent test conditions" for example, the same benefit-cost factors for the front, side and rear, it is therefore advisable to expend most effort on the improvement of the front structure.

Front

5 mph head-on fixed barrier impact, reversible
40 mph head-on fixed barrier impact, 0° or
60 mph head-on car-to-car collision
30 mph head-on car-to-car collision, 30°
30 mph head-on fixed 14 inch pole impact

Side

16 mph car-to-car collision, 90°

Rear

5 mph rear-end collision by rigid moving barrier, reversible
40 mph rear-end car-to-car collision

Rollover

As with passive safety, it is necessary to conduct benefit-cost analyses in the active safety field since equipping vehicles with automatic steering correction devices and anti-skid systems is very expensive indeed. Visibility requirements should likewise be measured on a benefit-cost basis. Unfortunately, no statistical reference material has become available for this work to date and without such data no analysis can be made. The handling properties stipulated in the ESV specifications do not take into account the capabilities of the human being in a total control system. No specifications should be laid down in this area as long as it is impossible to define good or bad handling properties. Otherwise, there is the danger that in this area the technological level would be classed as "good" only because, through coincidence, certain vehicles had certain properties.

FUTURE EVALUATIONS

The objectives for future tasks will be centered on the four following work sectors: Biomechanics, Vehicle Aggressivity, Vehicle Handling and Benefit/Cost considerations.

To us, biomechanics means finding the tolerance limits for the human body in typical traffic accident conditions and with commercial restraint systems.

One way of finding these tolerance limits is via statistical accident analyses which show the relationship between vehicle deceleration and/or impact speed and injury frequency and severity. Obtaining reliable correlation on the basis of statistical investigation is a very tedious and drawn out process.

A second possible method of finding the tolerance limits of the human body leads to tests on the human body. With this method it is not possible to avoid cadaver tests but voluntary tests are also considered. For some time now the Volkswagen company has been cooperating with research institutes in carrying out human tolerance tests with cadavers.

Frontal impacts at 50-mph equivalent barrier speed with the ESVW belt restraint system have already shown that no fatal injuries are to be expected in accidents with this system. It could also be proved in these tests that the danger of bending fracture to the vertebrae does not exist.

The knee bar as installed in the ESVW, together with the knee belt to restrain the lower part of the body was also examined for effectiveness with cadaver tests. This also supported our good dummy data since no injuries to the thigh or hip joint occurred.

A lot of scope remains for future biomechanical research in the investigation of impacts on hip joints and their reaction under different impact directions such as occur in accidents. We shall also concentrate our attention on the neck strength of the human body when it is restrained by advanced belt systems and on the impact resistance of the human brain as well.

The second aspect of our Experiments for Safer Vehicles programs is devoted to the field of vehicle aggressivity; that is, the problem which occurs in a collision between two vehicles of different masses and rigidity.

We divide the problem of vehicle aggressivity into two sections, first, the mass aggressivity which is evidenced by the fact that the large, heavy vehicle causes disproportionate deformation to the small vehicle, second, the structure aggressivity. The effect of the latter problem is shown by the fact that localized rigid parts of the structure penetrate the collision partner vehicle, or generally that different degrees of rigidity of the vehicles lead to severe localized intrusions.

In the development of the ESVW we have already demonstrated a theoretical possibility of how mass aggressivity can be reduced with hydraulic shock absorbers. This has shown that even in a collision between a passenger car and a truck the deformation of both vehicles does not exceed 40 inches (1 m), at an impact speed of up to 75 mph. There are, therefore, possibilities of preventing the major deformation from occurring on the small light vehicle in car-to-car collisions. We have already proved this in practical tests during the ESVW development, in that we carried out all car-to-car front, side and rear-end collisions with a 5,500 pound moving energy-absorbing barrier and did not exceed the tolerance limits for the occupants as laid down in the ESV Specifications.

This conception of mass aggressivity reduction with hydraulic shock absorbers is, however, an expensive solution and Volkswagen is working on a second concept to see if we can reduce mass aggressivity with purely mechanical and cheap deformation structures.

In the new ESV Specifications the problem of aggressivity is gone into for the first time and we are very pleased to see this. The specified tests with bogeys of different weights are presumably to prove the non-aggressivity of the ESV. We consider it necessary, however, to develop, within the framework of the ESV project, a general concept for non-aggressivity which takes into account the deformation characteristics of collision partners in future traffic conditions. On these lines, the proof of non-aggressivity means that

1. the deformation of the collision partners within acceptable limits is taken over by both partners, and
2. the vehicle decelerations of the collision partners in a large range of impact speeds do not exceed those values which occupants can survive, without injury, when equipped with advanced restraint systems.

The third feature of our Experiments for Safer Vehicles is directed towards increasing the active safety, in particular the handling problem. The old and also the new ESV Specifications contain a series of test regulations on this subject which are admittedly fulfilled by the ESVW and by nearly all our production vehicles as well but which, in our opinion, do not say anything about a vehicle with good or bad handling characteristics. There is a danger that the characteristics of a certain group of vehicles will be made the scale for assessment of good handling characteristics without actually taking into account the problem of matching driver and vehicle to one another. Volkswagen is devoting a lot of attention to this area by examining a large range of different vehicles and trying to find some correlation between subjective driver impressions and measured dynamic handling values. We consider vehicle simulation to be of great assistance in this type of test.

The Volkswagen ESV shows that even highly sophisticated technical requirements can be fulfilled. We think, however, that technical or quasi-technical requirements alone are not sufficient to solve the very complex problems of traffic accidents. Even when the ESV project is looked upon as a purely experimental project, the target surely cannot be to incorporate every measure which can make a contribution towards increasing safety, in the vehicle. The cost-effectiveness presents a limit here.

The benefit/cost analysis as a mathematical tool and used as an economic decision criterium is gaining, in its all-encompassing way, a special position in our program of special points in the Experiments for Safer Vehicles program.

This benefit/cost consideration tool appears to us to be an important daily aid for those of us who are working on serious technical and economical progress.

An important task in the future will be the preparation of statistical data on accident results which are the basis for solid benefit/cost considerations. As an example, a benefit/cost consideration of passive safety is explained here in more detail. The foundation of this work was the traffic accident analysis carried out in Europe by CCMC, which will be presented during this conference. This accident analysis defines the correlation between accident impact speed and the frequency of occupant injuries and deaths for the three main vehicle impact directions. With these accident statistics combined with the effectiveness of restraint systems, which is known approximately from tests, it is possible to estimate how high the expected reduction of injuries and fatalities would be if one were to design the vehicle structure for a definite test impact speed. The monetary benefit in this case is equivalent to the reduced number of injured and dead.

With the results of this benefit/cost analysis it is possible to make four main statements:

1. With a lap and shoulder belt usage rate of 15%, a benefit/cost factor of more than one can be obtained today without any improvement to the vehicle structure.
2. Every improvement to the vehicle structure in respect to increased impact test speed brings a benefit improvement in the form of reduced numbers of injured and dead, but the benefit/cost relationship drops steadily with the vehicle structure improvement because the costs rise out of proportion.
3. An improvement of the benefit/cost factor can be achieved if, based on the usual present-day vehicle structure, the usage quota of restraint systems alone can be increased to about 80%. This appears to be obtainable, for example, by making the wearing of belts compulsory or by means of a passive shoulder belt with knee bar.
4. An improvement to the vehicle structure, combined with a higher usage rate of the restraint system, brings the benefit/cost factor down, again. With this type of assessment it is quite clear that a vehicle which fulfills the requirements of the ESV Specifications is extremely far removed from the conditions where the expected benefit equals the costs which have to be expended.

The additional cost for structure, restraint system and their maintenance, alone, would add $2,000 to the cost of a normal medium class vehicle today. The new ESV Specifications do not bring any alterations in this respect either.

Just as it is possible to calculate a benefit/cost factor for the complete structure with restraint system, the front, side and rear structures can also be evaluated individually with the aid of this method, if the statistical data is also available for these impact directions.

It appears logical to us to distribute the costs for the vehicle structure improvements and the restraint systems in such a way that the benefit/cost relationship for these three main impact directions is the same. This means that the test speeds for frontal, side and rear end impacts must be so matched to one another that they are consistent with regard to the cost-effectiveness.

According to our calculations the following two combinations, among others, are consistent:

1. 50 mph frontal impact against fixed barrier
   20 mph car-to-car side impact
   35 mph car-to-car rear-end collision, or
2. 40 mph frontal impact against fixed barrier
   16 mph car-to-car side impact
   32 mph car-to-car rear-end collision

The reason for the considerably lower estimated side test speed given by us as consistent lies in the fact that the costs for the side structure are about 10 times higher in relation to the front structure with the same degree of energy absorption.

In comparison with this, the new specifications contain the combination

50/30/50.

After studying the new specifications the question arises as to whether the differences and the progress are sufficient, when compared with the first specifications, to encourage the firms to take new initiatives and to build a new Experimental Safety Vehicle.

We believe that with the list of objectives for the Experiments for Safer Vehicles program, we are drawing the correct conclusions from the knowledge gained with the first ESV project to enable us to commence with new and well-founded knowledge, on the construction of new cars on an economic basis acceptable to mass transportation.

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THE FEDERAL REPUBLIC OF GERMANY TECHNICAL PRESENTATION

SECTION **TWO** PART **2**

NEW INVESTIGATIONS OF HUK ACCIDENT RESEARCH: "INTERIOR SAFETY OF AUTOMOBILES"

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INTRODUCTION

Present Situation

In order to contribute actively towards an improvement in traffic safety, German automobile insurers started on detailed accident research in 1969. At the Third ESV Conference in Washington, we reported on the working methods and initial results of the study "Interior Safety of Automobiles."

Up to now, 92,000 accidents have been investigated by a team of engineers of the German Association of Third Party, Accident and Motor Traffic Insurers. 27,500 of these cases contained such detailed data on accident circumstances, damage to vehicles and passenger injuries that it was possible to subject them to a thorough, scientific analysis. By April, 1973, an additional 8,000 accidents will have been examined. By virtue of this interlocking succession of results achieved and by simultaneously analysing further accidents, it will be possible to carry out additional examinations on important problems quickly and in sufficient number.

Thus, additional investigations into injuries suffered in trucks, the influence of weather and road conditions on accident severity, and the benefit of an anti-locking system in an accident are now in the process of being carried out.

Benefit Of The Anti-locking System

From our investigation of the "benefit of an anti-locking system in actual accidents," a first report on 350 accidents can be submitted today. Work on this subject is fully started, so that in a few months we shall dispose of material on at least 2,000 accidents.

The problem of such an investigation lies in the requirement of producing figures indicating how much accidents could be reduced or prevented by the existence of an optimal braking system. It must also be clarified as to what importance ensues from the three effects of the anti-locking system; namely, stability of direction, maintenance of steerability and curtailment of braking distance.

Description Of The Supplementary Questionnaire

We have drawn up our own supplementary accident-injury sheet for these tests, in which information on the accident vehicles, their driving and collision speeds and the degree of damage to the vehicles — a measure for the severity of the accident — is recorded. The injuries suffered by the occupants in these accidents are specified. It can thus be clarified within what speed ranges and accident severity a particular benefit will be gained from improved braking systems.

The supplementary sheets also contain data on the driver and his driving experience and the vehicle, its age, its mileage and probable deficiencies of, for example, the tires. The nature of braking reaction and the length of the braking distance are indicated. The important influential road factors (e.g., type of roadway, road surface and route), as well as the environmental conditions (e.g., atmospheric and light conditions), are noted in detail. A report on the accident cause is also available in each case.

Test Material

The files of insurance companies are used as test material. They also serve as a basis for other accident investigations carried out by the German Association of Third Party, Accident and Motor Traffic Insurers. It should also be mentioned that we were concerned only with Third Party motor-liability cases; accidents involving other responsibilities were not analysed. There was thus no limitation on the results, due to the fact that the benefit of the anti-locking system in accidents with only one person involved can be deemed to correspond to those involving two vehicles. All accidents investigated occurred in 1969. We examined whether the 350 recorded accidents demonstrated a representative cross-section; Figure 1 shows the distribution by month. The tendency being, in the main, identical with the official federal statistics of 1969, is obvious. A comparison of the

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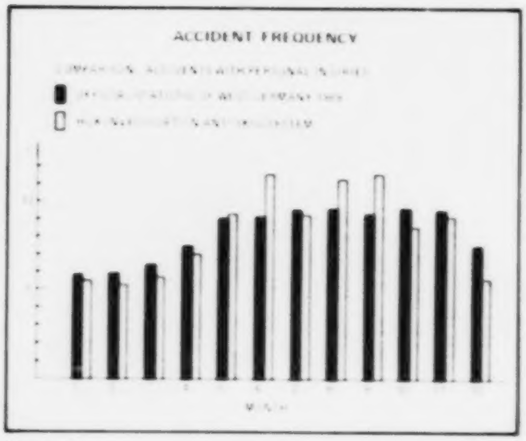


Figure 1

times of accident likewise showed full uniformity with our material of 10,271 accidents. For this reason, the hitherto existing investigation material corresponds to the normal distribution of accidents with personal injuries.

Classification Of The Anti-locking System In Benefit Categories

The question now is, to what degree can an anti-locking system be of benefit in accidents with injuries to passengers. In this respect, four categories have been created.

Category 1: No Benefit

Accidents in this category would not have been influenced by an anti-locking system. They involved accidents in which there was no braking reaction on the part of the driver, or where the accident situation occurred so unexpectedly that the accident was inevitable in spite of immediate optimal braking reaction.

Category 2: Benefit Possible

In these cases, the course of the accident, and thus the consequences thereof, would possibly have been favorably influenced by an anti-locking system if the driver's reaction had been optimal. The accident was, however, inevitable. Benefit from the anti-locking system cannot be excluded, but is, however, unlikely.

Category 3: Benefit Probable

It can be assumed that the accident in this category would probably have been prevented, or at least limited in its consequences, by an anti-locking system. However, one cannot as yet speak of a definitely ensured benefit.

Category 4: Benefit Certain

In these cases, the accident would certainly have been prevented by an anti-locking system. Although these accident divisions cannot be adhered to exactly in individual cases, practice has, however, proved that in general there has been no difficulty in allotting them to a group.

Figure 2 shows the allotment of 350 accidents to four benefit categories. These reveal the percentage of accidents with injuries to passengers in which no benefit from the anti-locking system was possible. In 70% of the accidents recorded by us, no braking reaction of the driver could be established.

In the 10% of accidents where the benefit of an anti-locking system seemed unlikely, but could not be completely excluded, the high ratio of rear-end collisions was particularly remarkable. Any beneficial effect of the anti-locking system would, in these cases, surely be conceivable only by means of a curtailment of the braking distance with simultaneous maintenance of steerability. This applies to accidents in intersection areas and in on-coming traffic where the benefit would, however, also depend on the behaviour of the other vehicle involved.

In the third category "benefit probable," rear-end collisions likewise dominate. In accidents involving "skidding through braking," it cannot be assumed with certainty that the accident would have been avoidable without the additional skidding at the moment of braking reaction.

In the category "benefit certain," accidents caused by skidding with blocked tires on straight stretches of road and in bends are preponderate. In these cases, the accident would certainly have been avoidable if

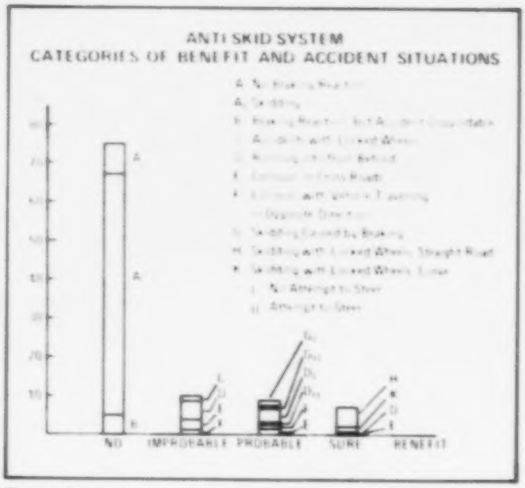


Figure 2

skidding had not occurred, because maintaining steerability was a decisive factor. To a certain degree, accidents caused by skidding on a straight road could have been avoided by improving the direction stability through an anti-locking system only on the rear axle, yet avoiding action was frequently still necessary in these accidents.

The division of the benefit categories according to accident situations clearly shows that, above all, importance must be attached to maintaining steerability and curtailing the braking distance on a wet roadway when evolving anti-locking systems.

In this connection, the influence of man must, however, be taken into account. In the cases tested by us, it was assumed that man derives benefit from the possibilities offered him by technology. A review of Category 3, accident situation G and H, shows that avoiding action would in many present-day accidents have been possible, but was not attempted. To what degree an anti-locking system stimulates the driver's reaction to taking avoiding action cannot, as yet, be said.

Furthermore, the question also arises as to whether the driver possibly does not nullify the improvement in the braking system by driving faster in bends or by keeping too short a distance to the car ahead. As a consequence of such driving behaviour, the real benefit from an anti-locking system could possibly be below the figures mentioned here. Independent of the technical development of the anti-locking systems, parallel investigations into driving behaviour should, therefore, also be carried out.

Possibility Of Determining The Equivalent Test Speed (ETS) From Actual Accidents

A decisive question for the evolution of vehicles and safety tests is how the velocities ascertained during investigations into real accidents involving the collision of vehicles coincide with the velocities ascertained in crash tests. Important conclusions could be drawn here with regard to the requisite crash conditions. Experimental results and the consequences of real accidents would thus be approximately comparable.

Determination Of Collision Speeds "Interior Safety"

In our tests, the collision speeds of both vehicles involved in an accident were estimated by our engineers from the insurance files. The collision speeds were then assigned to speed groups ranging from 0-30 k/mh, 31-60 k/mh, 60-100 k/mh and above 100 k/mh. In all these accidents, the relative movement of both vehicles towards each other was known to us. Furthermore, we were able to approximately establish the overlapping of the vehicles.

The tests were carried out on 2,400 head-on collisions involving 4,000 injured persons and converging speed directions of the vehicles involved. All accidents with insignificant overlapping of the vehicles were excluded. A speed gradation with increasing relative velocity of both vehicles was then formed on the basis of the collision classes. In determining the relative velocities, it was assumed that the great majority of all accidents of one velocity range was concentrated around the average speed between the upper and lower levels. Separate tests have shown us that this hypothesis is justifiable. Such simplification can be considered reliable, as the individual velocity classes embrace in part over 800 and 500 cases respectively.

After classifying all velocity classes in ascending sequence, the degree of damage to the vehicles, and the ratio of the severity of the injuries suffered by the occupants to the individual velocity classes, were recorded in percentages. The resultant curves, which ensued from the respective percentages, hardly point to any discontinuities and show a clear tendency for all types of accidents, thus verifying the gradation.

Figure 3 shows the velocity gradation at the moment of impact on the side surface. The higher ratio of deformation is clearly noticeable in the side-on collision vehicle - tree. (Velocity class "1/3".)

The relative velocity can be obtained from this figure, but not the quantity of collision energy. It can also be proved by these figures that other neglected factors, such as the weight of the vehicle, impact angle at the moment of collision, deformation energy and deformation behaviour of the vehicle structure, have no effect on the overall tendency. Although this influence would be quite remarkable if the number of

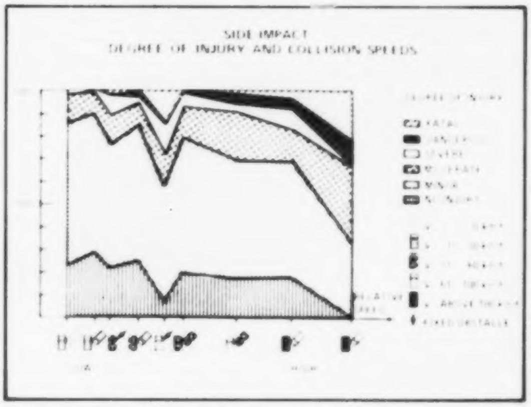


Figure 3

cases were smaller, one can with certainty speak here of a balancing-out of these respective factors in the accident figures under review. All data can be considered quasi as applying to a "European standard vehicle." From our investigations into the aggressiveness of vehicles, it follows that the influence of the vehicle weight does not always lead to a clear correlation as regards injuries. In addition to the factors so far mentioned, the influence of the "interior safety" of a vehicle on injury consequences is also initially neglected.

In the event of the total kinetic energy being transformed into deformation energy in a pure head-on collision involving two vehicles with complete overlapping, the bisected relative velocity of both vehicles would correspond to the Equivalent Test Speed (ETS) vehicle - fixed barrier. Here, the frequently differing vehicle mass and the varying deformation behavior of the vehicles would be neglected.

In a virtual head-on collision on the other hand, a certain proportion of kinetic energy is always transformed to deformation energy. The residual energy is consumed in the movement of the vehicles while slowing down after the accident. The damage to the vehicles and the injuries suffered by the occupants vary according to the extent of the ratio of deformation. To our knowledge, no conclusive results have so far existed as to what proportion of kinetic energy is transformed to deformation energy at the moment of collision. Criterions in this respect are supplied by crash-test pictures taken of a certain model at known impact velocities. In general, such pictures are, however, obtainable only from very few vehicle models at certain customary test velocities. In order to back-up our tests, we have also compared pictures of damages of this type with our cases of actual accidents. This method is not practicable for the great number of vehicles on the market, which have been recorded in our tests in a representative selection. By calculating precisely the collision processes in a sufficiently large number of cases, a direct relation between the energy of both accident vehicles before the collision, as well as the ratio of deformation energy and the movement of the vehicles at the moment of impact and thereafter, can be derived from the physical laws relating to accident mechanics. We made these calculations on 51 accidents.

The total collision energy was calculated from the respective mass of the accident vehicles and their speed at the moment of impact. The quantity of slowing-down energy could likewise be determined from the individual vehicle masses and their velocities. Thus, the energy gained, the ratio of deformation, as well as the ratio of slowing down, was known. It was then possible to calculate the percentage of deformation.

The ratio of deformation energy in head-on collisions in on-coming traffic can theoretically amount to 100%. Due to the never complete overlapping and the slowing-down movement of the vehicles resulting therefrom, this ratio is in practice in the region of 80%. In the majority of the cases involving head-on collisions, the deformation energy ranges between 50% and 60%. Collisions at intersections with injuries to occupants can be in the region of 40% - 50%, according to the nature of the collision. It must now be presumed that the ratio of deformation energy is transferred to both vehicles in equal proportion. But this hypothesis also does not prove true when a smaller number of cases is concerned. In the dimensions in excess of 1,000 accidents, however, the differing influences again balance out. Figure 4 shows the basic equation necessary for determining the "corrective factor deformation ratio:"

$$\frac{1}{2}\left(x_D \cdot \frac{m1}{2} v_1^2 + x_D \frac{m2}{2} v_2^2\right) =$$

$$\frac{m^*}{2} \mathrm{ETS}^2 \left(1 - e^{*2}\right)$$

$$\mathrm{ETS} = \frac{\frac{1}{2} x_D \cdot \mathrm{Eges}}{\frac{m^*}{2}\left(1 - e^{*2}\right)}$$

m^* = mass of the respective vehicle for calculation of the ETS

e^* = impact factor of the respective vehicle = f (v)

x_D = deformation ratio to total energy

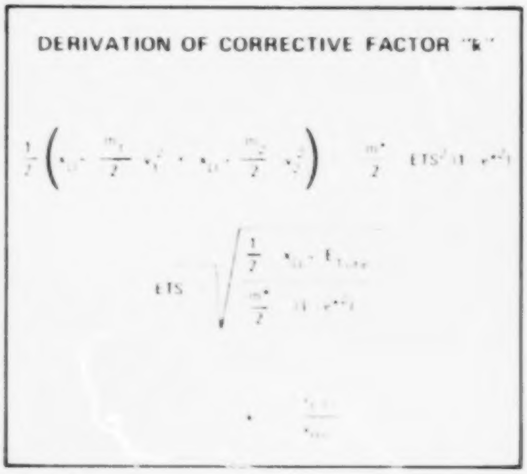
Eges = total kinetic energy immediately before the collision

$m_{1/2}$ = vehicle mass

$v_{1/2}$ = vehicle velocity before the collision

In each case, the ETS is calculated for both vehicles in this statement, whereby the influence of the differing vehicle mass is taken into account. The influence of the respective vehicle-deformation characteristics and their velocity dependency can also be included in the calculation. The respective ETS values results therefrom.

On the basis of the ascertained deformation ratio x_D to the total collision energy, the ETS is now



DERIVATION OF CORRECTIVE FACTOR "k"

Figure 4

calculated by way of the equations (Figure 4). From the ratio between the relative velocity of the accident vehicles, which we determined shortly before the collision, and the ETS, now established, the corrective factor $K = \frac{v\,ETS}{v\,rel.}$ is calculated. Due to the influence of mass and also the impact factor, this corrective factor is not the same as the deformation ratio.

The calculation of the corrective factor K is shown in Figure 5. In actual traffic accidents, an almost linear dependency ensues between the relative velocity of the vehicles and the ETS of a vehicle upon impact with a fixed barrier. The spread of the values is extremely small and can lead to a maximum tolerance of plus minus 5 km/h. This figure also embraces the results from the experimental tests on model vehicles carried out by Dr. Hagen at the Technical University of Munich in 1965. In these tests, model vehicles with deformation characteristics corresponding to actual vehicles with variable degrees of overlapping and differing speeds were induced to collide. In these cases, the ratio of deformation energy and the corrective factor vis-a-vis the ETS in head-on collisions were also calculated. The results of the experimental tests and the calculations of actual accidents coincided so closely that one can speak of absolutely reliable values as regards the corrective factors applied here.

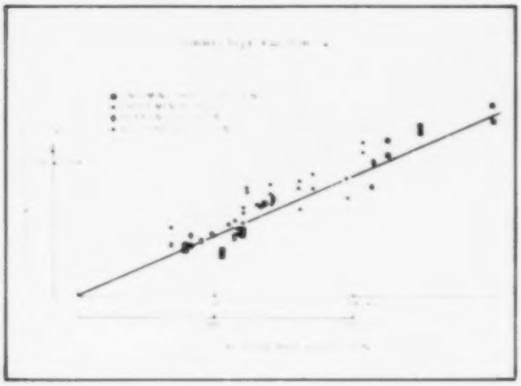


Figure 5

Thus, it will be possible for the first time to allot the collision class, which had so far been classified in accordance with increasing speed only, to an Equivalent Test Speed (ETS). This method can be applied with respect to all types of accidents. Hence, a direct dimension ensues between the collision energies occurring at collision and the resultant injuries suffered by the occupants.

Figure 6 shows this connection in a head-on collision with converging speed directions of both vehicles involved. The resultant injuries of all occupants are recorded in percentages for the ETS velocity intervals of 10 km/h in each case. The injury index corresponds to the classification applied in the accident tests carried out by the Cornell Aeronautical Laboratory and is practically identical with the "AIS Scale" of the NATO tests now being carried out in various countries. Injury degree "1" comprises minor injuries. Injury degree "2" – moderate – is sharply separated as to the severity of injuries, so that considerable injuries cannot be falsely interpreted. Injury degree "3" – severe – comprises severe injuries, such as compound fractures of the thigh or fractures of the skull where, however, no danger to life is involved. In injury degree "4" – dangerous, survival probable – and injury degree "5" – critical, survival doubtful – there is danger to life. The injury degrees

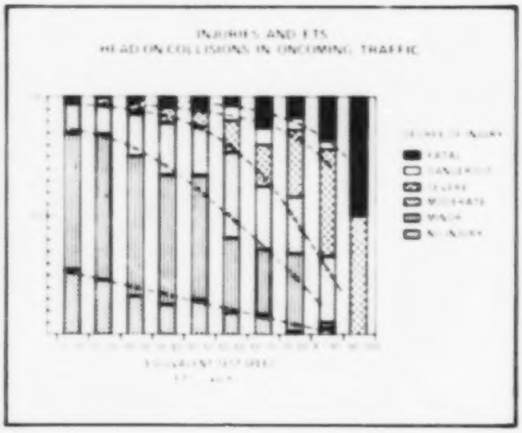


Figure 6

"6" and "7" comprise fatal injuries to one or more parts of the body.

In Figure 6, the number of persons remaining uninjured declines very rapidly with increasing velocity. It also proves, however, that in real accidents persons can, by fortunate circumstances, remain uninjured in the vehicle even at very high collision speeds. At relatively low collision speeds, there are far more slightly-injured persons involved. Even at an ETS commencing at 10 km/h, however, we ascertained a percentage of 15% of all occupants with considerable injuries. From 50 km/h upwards, half of the occupants suffered considerable injuries.

A particularly encumbering factor in assessing the safety of all vehicles on the market today is that dangerous and fatal injuries have also been established in all sectors of the ETS from 10 km/h upwards. The percentage of 2% may at first appear to be small. Due to the fact, however, that the ratio of injured persons has been applied to the sum of all occupants within the speed group under consideration, the death ratio of 2 percent is still equivalent to 18 persons killed.

In conclusion, this figure shows very clearly that there is a considerable risk of severe injuries at low ETS. This risk could be decisively reduced by utilizing all safety measures available today.

The question now arises as to what ETS ranges vehicle safety must be improved to achieve some benefit, at justifiable expenditure, for the great majority of persons injured in traffic accidents. The frequency curves of injury severity are shown in Figure 7 and have been allotted to the various ETS ranges. For classification purposes, we have combined the groups of uninjured persons, persons not dangerously injured (injury degrees 1 - 3) and dangerously and fatally-injured persons (injury degrees 4 - 7). The focal point, as regards uninjured persons, lies in the ETS range of 10 - 40 km/h. For the total of all persons injured only an insignificant additional benefit ensues from safety measures in head-on collisions from 60 km/h upwards. In this range, the percentage of serious injuries is, however, very high.

Upon reviewing the distribution of frequency with regard to non-dangerous injuries, it must be pointed out that from an ETS of 25 km/h upwards, a quarter of these injuries is "moderate" or "serious." Dangerous and even fatal injuries already occur from an ETS of 15 km/h. Upwards of an ETS of 30 km/h, the percentage of dangerously and fatally-injured persons remains practically constant in all subsequent speed ranges, in terms of figures. As Figure 6 has already demonstrated, the ratio of fatally-injured persons increases, however, quite considerably with increasing velocity.

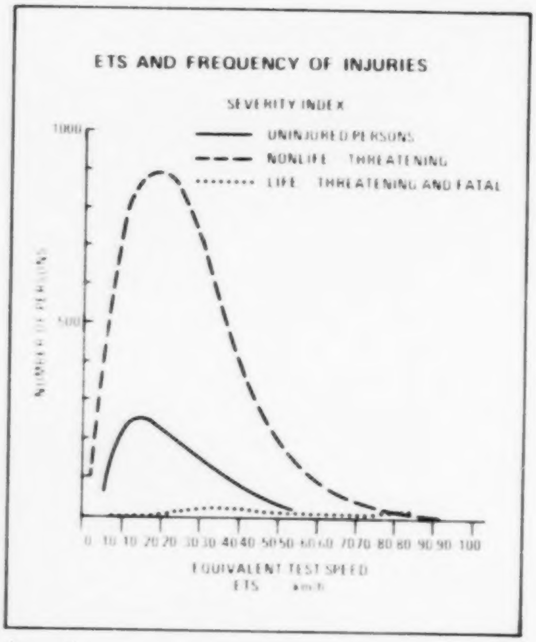


Figure 7

Conclusions To Be Drawn For The Effecting Of Safety Tests On Vehicles

The method of allotting actual traffic accidents with head-on collisions to an Equivalent Test Speed (ETS), as described by us, embraces the most frequent type of accident occurring in practice. As former studies have already shown, approximately 60% of all vehicles sustain damage to the front end in accidents involving personal injury. Overlapping of the complete front-end surface in front-end vehicle tests against a fixed barrier only seldom occurs in an actual accident. Collisions with one-sided overlapping predominate. Approximately 60% of the energy ensuing is transformed to deformation energy at converging speed directions of the vehicles involved. Present-day test conditions should perhaps be adapted, to a greater degree, to the conditions existing in practice. By virtue of the other relative-velocity direction in accidents at intersections with side surface damage, the ratio of deformation in these accidents is less high, being in the region of approximately 40% - 55%. The focal point, as regards injury frequency, lies between 10 and 40 km/h ETS. The preponderate part of dangerously and fatally-injured persons also accrues to the range between 25 and 60 km/h.

Considerable improvement in the "Interior Safety" of vehicles is, above all, necessary up to an ETS of 60 km/h, in which range more than 90% of all types of injuries would be embraced. These improvements could largely be achieved with the means already available today. Restraint devices for the occupants of front seats assume a major role in this connection. These restraint devices must be designed in such a way that they ensure — partly by easy operation, partly by passive systems — a high degree of effectiveness and a high quota of actual use. Definite results are certainly to be anticipated from these measures in the relatively seldom ETS range in excess of 60 km/h.

By means of this method which, to our knowledge, has been applied for the first time, it has now become possible to draw conclusions with regard to the ETS in vehicle crash tests against a wall barrier by the determination of actual collision speeds, provided that a sufficiently greater number of investigated collisions is available. Theoretical tests, as well as experimental results and exact calculations in respect of actual accidents, have served as a basis for these considerations. We have also compared the test results of other countries available to us with our own results and have arrived at favorable comparisons. By virtue of this method, the categorization of actual accidents to experimental results is possible. The actual road traffic conditions existing can be reconstructed. The results presented in this connection could, for this reason, constitute a contribution towards intensification of the benefit/cost analyses relating to practical experience. They should also contribute towards recognizing more clearly the scope of validity of such experiments. The research results thereby obtained clearly show that the demand for an ETS of less than or equal to 60 km/h ≙ 37.5mph suffices.

The German Association of Third Party, Accident and Motor Traffic Insurers will, in all likelihood, continue their past tests in the years ahead. We shall then also have at our disposal the corresponding accident figures to enable us to apply this method to the detailed problems of vehicle construction and occupant injuries. It is our aim to prove and underline the efficiency of traffic safety measures in actual accidents and we thereby hope to contribute in this way to solving the question of seeking out real possibilities for improving vehicle safety in the future.

THE FRENCH TECHNICAL PRESENTATION

SECTION TWO PART 3

INTRODUCTION

MICHEL FRYBOURG, *Director*
Institute for Transportation Research
Ministry of Transport

It is my pleasure to introduce the French Technical Presentation on our ESV work. First, I will present a paper on "Traffic Safety Comparison of Various Classes of Common French Car Makes and Drivers" authored by Mr. Berlioz, Mr. Baylatry and Mr. Filou. Today's agenda lists the other presentations which will follow.

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THE FRENCH TECHNICAL PRESENTATION

SECTION TWO

PART 3

TRAFFIC SAFETY COMPARISON OF VARIOUS CLASSES OF COMMON FRENCH CAR MAKES AND DRIVERS

C. BERLIOZ,
B. BAYLATRY
and
C. FILOU
Presented by
MICHEL FRYBOURG, *Director*
Institute of Transportation Research
Ministry of Transport

At the last conference in Washington, we showed that cars which are the most vulnerable in crashes into other vehicles are less vulnerable when crashing into fixed obstacles on the same type of road.

The previous study dealt with post-crash safety. Here, we intend to set out the most global results, including the pre-crash and post-crash safety, by analyzing the safety in terms of the rates of victims related to the distance covered.

We try to make a judgment, not of the intrinsic qualities of such and such class of vehicles but, of the associated pairs (vehicle-driver), by taking into account both the age of the driver and the type of vehicle.

Figure 1 shows the different associated pairs that are distinguished in this study. The partitioning results from two decompositions, the one into seven classes of cars, the other into two classes of age. For convenience, we denote all drivers under 25 as young, and all drivers of 25 and over as elderly.

The safety is measured on the one hand by the involvement rate in the accidents, that is to say, by the frequency of accidents related to the distance covered, and on the other hand by the crash-gravity rate, that is to say, by the number of casualties (killed or seriously injured) related to the accidents and finally by the frequency of casualties related to the distance covered.

The basic data used to estimate the travels made by different classes of vehicle drivers on rural highways outside built-up areas are relative to 1969, as it is the same for those of personal injured accidents.

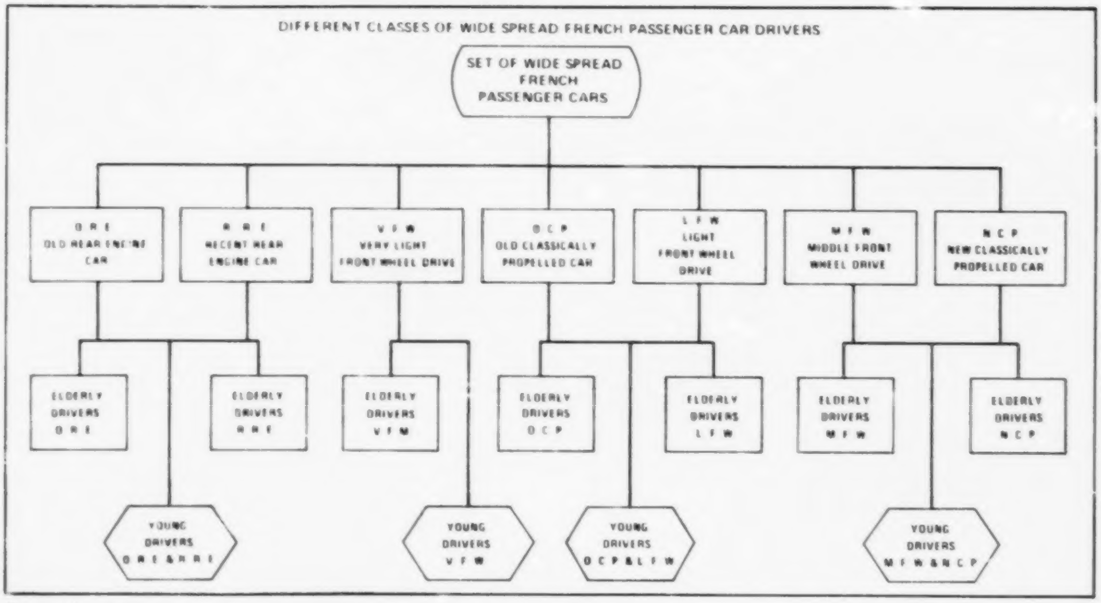
DIFFERENT CLASSES OF WIDE SPREAD FRENCH PASSENGER CAR DRIVERS
SET OF WIDE SPREAD FRENCH PASSENGER CARS
O R E OLD REAR ENGINE CAR
R R E RECENT REAR ENGINE CAR
V F W VERY LIGHT FRONT WHEEL DRIVE
O C P OLD CLASSICALLY PROPELLED CAR
L F W LIGHT FRONT WHEEL DRIVE
M F W MIDDLE FRONT WHEEL DRIVE
N C P NEW CLASSICALLY PROPELLED CAR
ELDERLY DRIVERS O R E
ELDERLY DRIVERS R R E
ELDERLY DRIVERS V F W
ELDERLY DRIVERS O C P
ELDERLY DRIVERS L F W
ELDERLY DRIVERS M F W
ELDERLY DRIVERS N C P
YOUNG DRIVERS O R E & R R E
YOUNG DRIVERS V F W
YOUNG DRIVERS O C P & L F W
YOUNG DRIVERS M F W & N C P

Figure 1

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In Figure 2, we can notice the effect of the age of the drivers upon the relative involvement risks of different types of vehicles related to the distance covered. According to the earlier results achieved in France, as well as in foreign countries, the relative involvement risks of young drivers are high. Nevertheless, we disclose here another, much more important, result — the multiplying factor of the risk of young drivers in comparison with the elderly drivers is not uniform at all. If this factor is at the rate of 1.3 for the very light front wheel drives (V.F.W.), it is at the rate of 2 for the rear engined cars (O.R.E. + R.R.E.); at the rate of 2.5 for the heaviest and fastest cars (M.F.W. + N.C.R.).

Thus, there is a reciprocal action aggravating the risk, on the one hand, between young drivers and the rear-engined cars, on the other hand, between young drivers and the vehicle top speed.

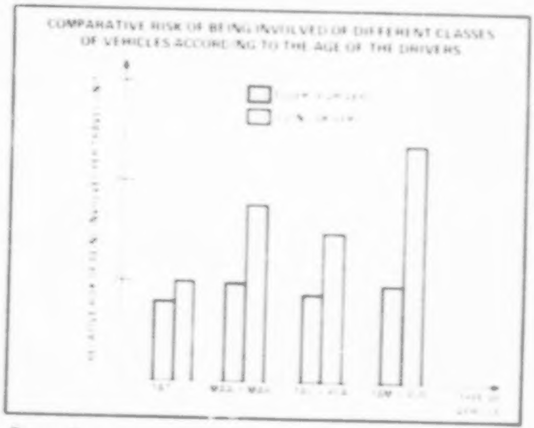

Figure 2

Figure 3 shows the relative involvement risk per travel unit of elderly drivers, varying with the average top speed of different classes of vehicles.

We can state that, if we put apart the rear-engined cars, the involvement risk of elderly drivers will increase, together with the top speed, considerably and regularly, despite the criterions of design and the more or less recent conception of the vehicle. In a more precise way, an increase of 10 km/h will lead to the absolute value of the risk: an increase of 3%.

The results given now take into account the gravity of the accidents. In order to eliminate a possible effect upon the occupancy rate of vehicles, only drivers and pedestrians will be taken into consideration.

Having care for a better differentiation between the aggressiveness and a protection of a given pair vehicle-driver, we have distinguished among the driver victims, between inner injured drivers and outer injured drivers.

For a given class of vehicles, we understand:

- Outer injured drivers: all drivers and pedestrians who are killed or seriously injured in accidents involving a vehicle of a given class, but not driving a car of this class.
- Inner injured drivers: all drivers who are killed or seriously injured, driving a car of a given class.

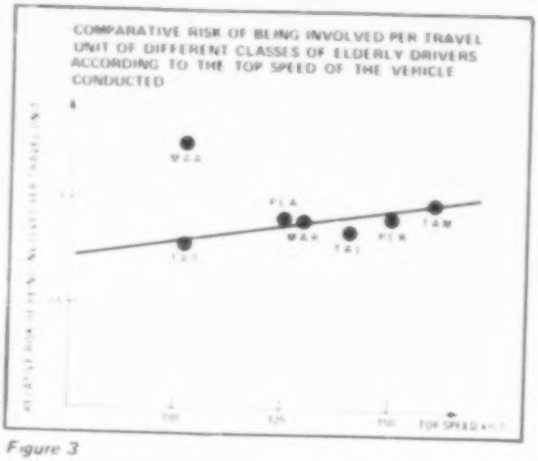

Figure 3

Figure 4 shows the relative risk, per travel unit, of serious injuries inflicted on outer drivers, varying with the average top speed of different classes of vehicles.

We notice a great and regular increase of the crash gravity, together with the top speed when the driver is more than 25 years old. Consequently, for a top speed increase of 10 km/h, the rate of outer injured drivers related to the distance covered increases 12% in absolute value.

We are presented with two phenomena going in the same direction:

- The slowest and lightest classes of vehicles are the less aggressive in the accidents.
- These classes that we have previously examined, are also the less frequently involved in the accidents.

In Figure 5, we can examine the relative risk per travel unit of inner injured drivers, varying with the average top speed of different classes of vehicles. It appears quite clearly that the risk for the elderly drivers is practically independent of the average top speed of different classes of vehicles.

Here, two phenomena compensate each other:

- The slowest and lightest classes of vehicles are less frequently involved in the accidents.

- These classes secure, on the whole, not as much protection during the crash, though the drivers are less seriously injured in crashes into a fixed obstacle. Furthermore, as shown in Figure 4, a systematic aggravation, together with the top speed, of comparative performances of drivers under 25 in comparison with the elder drivers.

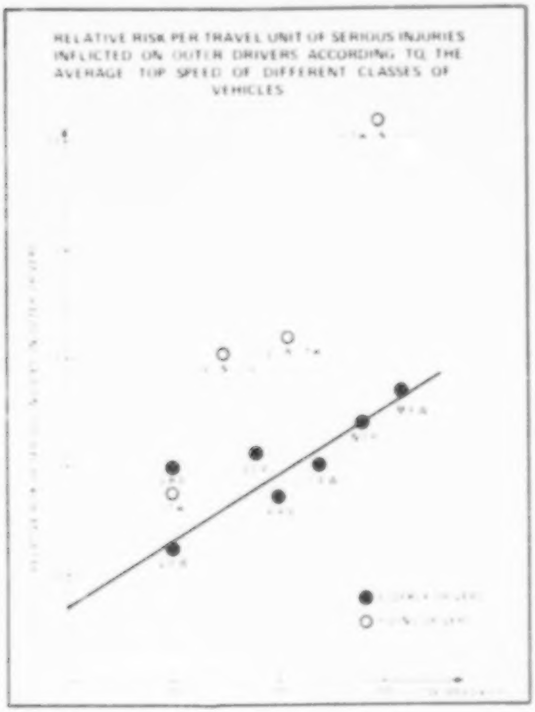

Figure 4

Figure 6 shows the synthesis of the two preceding; it shows the total risk, per travel unit, of serious injuries inflicted on drivers. Thus, for elderly drivers, it appears that the total risk increases 6% in absolute value for a top speed increase of 10 km/h.

In France, there are important differences between various groups of drivers concerning the frequency of drivers involved per travel unit in personal injury accidents, as well as its severity. These results are already known by French insurance companies, in terms of the annual frequency and its cost, the influence of the vehicle speed is primary to safety. We notice that, except for older vehicles, the frequency of injured drivers increases about 6% for each 10 km/h top speed increase.

We have described the French situation, which certainly is particular, due to the structure of the vehicles, the nature of the road system, and the driver's state of mind, to whom speed often has a social value, and who do not respect the speed limitation established on a certain number of roads (in 1969, it was on 1600 km). The situation goes

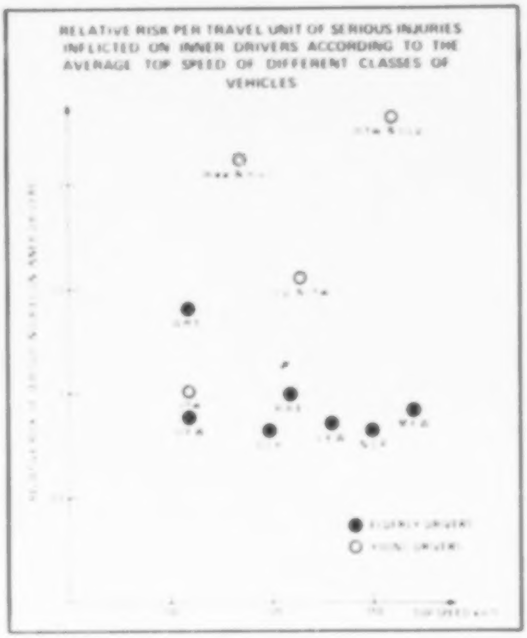

Figure 5

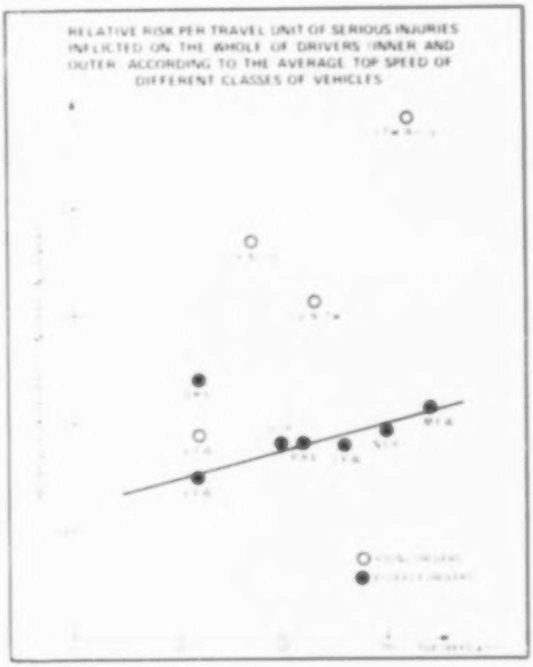

Figure 6

beyond its national framework, by proving that safety improvement is a complicated task which requires a system-analysis approach, in which interactions between subsystems cannot be ignored. Despite the intrinsic qualities of roadholding and crashworthiness, the heaviest and fastest cars have the highest accident and injury rates, due to the way the drivers use their vehicles.

A regulation prescribing the same specifications for all types of vehicles seems to be dangerous for France, for it could eliminate the V.F.W. class, which seems very safe.

THE FRENCH TECHNICAL PRESENTATION

SECTION **TWO** PART **3**

A STUDY OF ANTI-LOCKING BRAKING SYSTEMS

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OBJECT AND SCOPE OF THE STUDY

The purpose of this study was to try to determine, *a priori*, the effectiveness of the anti-locking braking system, i.e., by analyzing accident situations from files gathered during the course of the VEHITEST study (effectiveness of periodic technical verification of vehicles).

Collection of the sample was carried out in seven French departments from May, 1970 to June, 1971. 1,121 accident files were studied.

Study Method

The following information was available to us:

Type of accident

1. Accident involving a single car (loss of control)
2. Accident involving a car and at least one motorcyclist, cyclist or pedestrian
3. Accident involving at least two vehicles (car or truck) at an intersection (without overtaking)
4. Accident involving at least two vehicles (car or truck) after passing
5. Accident involving one vehicle stopped on the roadway (car or truck), or two vehicles travelling in the same direction, not involving passing and not at an intersection (reduction in speed)
6. Accident involving at least two vehicles (car or truck) travelling in opposite directions (not involving passing and not at an intersection)
7. Others

Condition of the road surface following inclemencies: dry, wet, snowy, icy ground.

Make and type of vehicle involved in the accident.

The advantages of an anti-locking system are:

- eliminates braking instabilities, especially swerves
- obtains a better braking distribution and maximizes it as a function of the distribution of the load
- eliminates braking errors
- makes braking possible and virtually safe in certain driving situations — around curves, on wet, snowy or icy pavement
- mainly, maintains driveability and stability of the braking vehicle, without noticeable increase in braking distances. (Possible gains in braking distance were not taken into consideration).

It was possible, using an *a priori* estimate, to investigate cases where there was a reasonable probability that equipping vehicles with such a system would have prevented the accident. Different cases were studied where wheel-locking occurred. We tried to determine whether a preventive maneuver, made possible by using an anti-locking system, could have swerving or loss of control following excessive braking. The coefficients of subjective probability of suppression for each accident can be determined, provided the driver remains calm and is capable of reacting, that is, if he can perform the preventive maneuvers necessary with a vehicle equipped with an anti-locking system. A few characteristic cases are given in Appendix 3. It is certain that the actual behavior of the driver will not always adapt itself to the circumstances. For an idea of this adjustment each accident was studied in accordance with the following classification:

- *Correct behavior*: the preventive attempt was proper under the circumstances and with the information available to the driver.
- *Incorrect behavior*: often a maneuver made as a reflex action, without considering the actual situation (e.g., preventive maneuver in the direction of the pedestrian crossing the road).
- *Passive behavior*: the driver braked; no maneuver was attempted.
- No maneuver was possible or necessary.

The results obtained with this classification permitted the formulation of hypotheses for the adaptation of maneuvers which can actually be carried out by drivers involved in accidents, if their vehicles are equipped with an anti-locking system.

For each accident where the anti-locking device might have been useful, a coefficient of subjective probability of suppression of this accident was estimated between 0.1 and 1. (In the Tables the coefficients are grouped in three classes: 0.1 to 0.3; 0.4 to 0.6; 0.7 to 1).

The total percentages of accident suppression, calculated with the above coefficients, will be called "proportion of avoidable accidents" for the remainder of this study. After the application of a coefficient of maneuver adaptation, (estimated with the study of maneuvers actually performed in these accidents as a base), the percentages will be referred to as the "proportion of avoided accidents."

COLLECTION AND PROCESSING OF DATA

The Poll

- *Principle of the poll*

  The polling plan is that of the VEHITEST investigation.

Two thousand accidents were studied in eight departments corresponding to eight geographical sectors. Four centers worked in conjunction with the Gendarmerie Nationale and the four others with the Surete Nationale. The sampling retained in each department allowed a sampling structure to be obtained which was similar to the structure of bodily accidents for France as a whole. The following four variables were used:

- Distribution of accidents between the Gendarmerie Nationale and the Surete Nationale
- Seriousness of accidents
- Distribution of accidents in and outside of an intersection

Details on the distribution of the accidents to be studied are given in Appendix 1.

- Selection of accidents

The files used dealt only with accidents involving touring cars and small trucks, hereinafter referred to as L.V. (light vehicles).

These accidents were divided into four classes:

Accident occurring: during a work day — during a Saturday, Sunday or holiday.

Accident occurring: during the day — at night.

This division enabled "special service" bulletin boards to be set up.

Accidents reported during these 12-hour special-duty periods (7 A.M. – 7 P.M. and 7 P.M. – 7 A.M.) were selected for the study and files were forwarded by police or law enforcement agencies.

The number of special-duty night workers was lower for practical problems than the number of special-duty workers needed in order to respect the distribution of bodily accidents in France with regard to general accident statistics. Not every accident report could be collected, and the Surete Nationale sample is underestimated. Results will have to be weighed as a function of Surete Nationale and Gendarmerie Nationale distributions and day-night distributions.

Nature of the Data

The accident reports (Appendix 2) were the primary source of information; they included: a map of the location, photographs, and declarations made by the persons involved and by witnesses.

Rectifying the Sample

The accident sample was weighed so as to obtain a sample structure similar to that of the structure for bodily accidents in France.

The following criteria were employed:

- Control agency (Surete Nationale – Gendarmerie Nationale)
- Day – Night

With the following symbology:

N — Population of accidents in France

N_i — Number of accidents in France in category i

$$P_i = \frac{N_i}{N}$$

n — Accident sample

n_i — Number of accidents for category i in the sample

$$p_i = \frac{n_i}{n}$$

If the sample were correct, we would have $P_i = p_i$

To re-establish this equality, the required weight will be such that

$$\alpha np_i = P_i \text{ or } \alpha_i = \frac{nP_i}{n_i}$$

The coefficients α_i obtained are:

Gendarmerie Nationale	Day	0.48
	Night	1.14
Surete Nationale	Day	1.42
	Night	6.43

Calculating Method Of The Estimated Safety Advantages

The calculation was accomplished by applying the coefficients of subjective probability of suppression of the accident and the corrective coefficients relative to maneuver adaptation, to the accidents where anti-locking seemed necessary.

$$\text{Percentage of avoided accidents} = \frac{100\,K}{N}$$

$$K = \sum_{i=1}^{N} A_i\, k_i$$

N = Accident sample (conclusive surveys)

A_i = Accident i requiring the use of the anti-locking system

k_i = Coefficient of subjective probability assigned to accident i. ki can take on the values 0.1 to 1

K = Corrective coefficient of adaptation of maneuvers corresponding to the hypothesis (K_1, K_2)

Calculating Method Of The Cost-Advantage Of An Anti-Locking System

For all vehicles

The admissible increase in the price of the vehicle for safety installations is calculated in Appendix 4.

A corrective measure which procures a decrease of 1% for accidents occurring with private automobiles, with no change in the seriousness of the accident, must, if it is to be "paying," entail a per-vehicle increase in a pre-tax price lower than 49 Francs.

The maximum admissible cost of the anti-locking device, corresponding to a given advantage/cost ratio, was calculated in the following manner:

maximum admissible cost of the anti-locking system

$$= \frac{\%\text{ of decrease} \times 49\text{ F}}{\text{advantage/cost ratio}}$$

For a certain category of vehicles

The maximum 49-Franc increase in the pre-tax price of the vehicle obtaining a 1% decrease in accidents is an average value. For each category of vehicle the risk of involvement and the quantity of accidents where vehicles of this category are involved may differ. The coefficient of relative risk, equal to the product of the risk of involvement by the rate of seriousness, allows the calculation of the value of the admissible increase in pre-tax price of the vehicle for a given category. This coefficient of relative risk was studied in the *Study of Comparative Safety for Different Classes of Touring Cars*.

The maximum admissible cost of the anti-locking system for a category of vehicle is equal to:

$$\frac{\%\text{ of decrease in accidents for the category in question} \times 49\text{ F} \times \text{coefficient of relative risk}}{\text{advantage/cost ratio}}$$

The calculation is approximate, since for the coefficient of relative risk, the rate of seriousness equal to the

$$\frac{\text{number of killed} + \text{number of seriously injured}}{\text{total number of accidents}}$$

represents an average rate of seriousness in accidents on national highways outside of urban agglomerations.

ANALYSIS OF THE RESULTS

Frequency Of Accidents In Which The Anti-locking System Might Have Proved Useful (Table II)

From the 1,121 accident files examined, we concluded in 391 cases (i.e., 35% of the accidents) that an anti-locking system might have been useful and would have in all probability prevented the accident.

No conclusion was reached for 75 files, (6.7% of the accidents) since information available to us was incomplete or the circumstances surrounding the accident were difficult to determine.

Study Of The Adaptation Of Preventive Maneuvers

The different types of maneuvers are defined in Table I. These maneuvers are categorized as follows:

- In 16% of the accidents the maneuvers were correct for the situation
- In 10% of the accidents the maneuvers were incorrect for the situation
- In 50% of the accidents no maneuver was undertaken
- In 11% of the accidents no preventive maneuver was necessary

Table I shows the different types of maneuvers, and the distribution of accidents in which an anti-locking device might have proved useful, as a function of the coefficients of subjective probability of suppression of the accident.

The X^2 homogeneity test, carried out on the distribution obtained by grouping two categories of coefficients of subjective probability, 0.1 to 0.3 and 0.4 to 0.6, was shown to be significant at the threshold of 5%:

- For incorrect maneuvers, the coefficients of subjective probability of suppression of the accident are higher than for the cases as a whole.
- The accidents for which the preventive maneuver performed is adapted to the circumstances and the accidents requiring no special maneuver have a distribution comparable to that of the maneuvers as a whole.

Since the study method of the accident files was such that the coefficient of subjective probability of suppression of the accident was determined by admitting that the maneuvers performed were adapted to the circumstances, the coefficient had to be corrected by a coefficient of maneuver adaptation.

Hypotheses relative to this corrective coefficient can, by reason of their approximative nature, be formulated, even though prior results are non-homogeneous.

Low hypothesis: In 30% of the accidents where an anti-locking device might have been useful, the possibilities offered by this device were utilized correctly by the driver and enabled the decreases in accidents predicted when the files were studied.

This hypothesis stems directly from the observation of the present situation: the preventive maneuvers were correct – or – no maneuver was needed in 27% of the cases.

High hypothesis: The predicted decreases in accidents were possible in 60% of the cases. Here we pre-supposed that a part of the passive or incorrect behavior of the driver arose from his stupification when faced with the unexpected reactions of his wheel-locked vehicle, or from a defeatist attitude which could be mitigated by confidence in the device.

Estimated Safety Advantages

Frequency of avoided accidents.

The following are the results:

Low hypothesis: 6.7% of the accidents could be avoided by equipping all moving vehicles with an anti-locking system. The symmetric confidence interval having a probability equal to 0.95 of overlapping the true value of the frequency is 5.2% - 8.2%.

High hypothesis: 13.4% of the accidents could be avoided: confidence interval: 11.3% - 15.5%.

Variation Of The Frequencies Of Avoidable Accidents As A Function Of The Type Of Accident And Of The Condition Of The Road Surface

- Variations by type of accident

Table III gives the distribution of avoidable accidents (in a theoretical hypothesis of 100% correct behavior) by type of accident. The X^2 test showed itself to be significant at the threshold of 1%.

- The proportion of accidents avoidable by using an anti-locking system is greater for accidents involving a vehicle stopped on the roadway or a decrease in speed, than for all the cases.
- Accidents of the second type, involving a two-wheel vehicle or a pedestrian, have a proportion of avoidable accidents which is lower than that for all the cases.

The total percentages of decrease in accidents are given by type and as a function of the hypotheses of adaptation of maneuvers in Table IV. These values confirm the preceding results.

- Variations as a function of the condition of the roadway surface

Table VI gives the distribution of avoidable accidents as a function of the condition of the roadway surface. The X^2 test showed itself to be significant at the threshold of 5%.

- The proportion of accidents avoidable through the use of the anti-locking system is greater on wet, snowy or icy roads than on a dry road, but this divergence is smaller than what might have been expected.

Table V shows that an anti-locking system is needed one out of every three times on a dry road and one out of two times on wet, snowy or icy roads. The X^2 test relative to Table VII showed itself to be significant and indicates that the coefficients of subjective probability of suppression of the accident are less for a wet, snowy or icy roadway than for a dry one.

The study method for the accident files probably led us to underestimate the importance of the effect of an anti-locking device on a slippery road surface; our information for this case is the least complete of all: too few indications on braking and locking tracks.

Variation Of The Frequency Of Avoidable Accidents As A Function Of The Categories Of Vehicles Equipped With The System

Based on weight class, maximum speed, and type of propulsion, the following categories of vehicles have been defined:

French automobiles in wide use:

a. Vehicles no longer being manufactured (as of 1971)

Weight from 560 to 795 Kg – maximum speed 95 to 135 km/h

b. Low-range vehicles, still manufactured (in this category 7% are rear-drive vehicles and 93% are front-wheel drive vehicles)

Weight 495 to 725 Kg – Maximum speed 85 to 125 km/h

c. Middle-range vehicles, front-wheel drive (all still being manufactured)

Weight 800 to 940 Kg – Maximum speed 138 to 145 km/h

d. High-range vehicles, front-wheel drive (all still being manufactured)

Weight 980 to 1,400 Kg – Maximum speed 140 to 175 km/h

e. High-range vehicles, traditional drive (still being manufactured)

Weight 900 to 1,200 Kg   Maximum speed 130 to 160 km/h

- Other French cars of limited use and motor-vans
- Foreign cars

Table IX gives the distribution of avoidable accidents for these different categories of vehicles. The distribution of avoidable accidents differs significantly according to the makes and types of vehicles involved.

For high-range, front-wheel drive vehicles the percentage of accident decrease is significantly higher than for the other categories. This type of result is maintained for the following groupings:

- High-range vehicles
- Middle and high-range front-wheel drive vehicles

For low-range vehicles the percentage of accident decrease is significantly lower than for the other categories.

COST-ADVANTAGE OF THE ANTI-LOCKING SYSTEM

In the different hypotheses of maneuver adaptation, the maximum admissible costs of an anti-locking system are given in the following:

	Hypotheses of maneuver adaptation	
	Low hypothesis	High hypothesis
Installation in all cars	340 F	660 F
Installation in high-range cars	450 F	890 F
Installation in low-range cars	185 F	370 F

The advantages obtained with an anti-locking system are maximum when the device is installed in high-range vehicles. For this category of vehicles the percentage of avoidable accidents and the coefficient of relative risk are higher.

For low-range vehicles the percentage of accident decrease and the coefficient of relative risk are low.

So that the advantage-cost ratio approaches 1, in a hypothesis where all vehicles are equipped with the system, the pre-tax price of the anti-locking system would have to fall between 340 and 660 F. This supposes mass production and more economical solutions than those currently proposed.

CONCLUSION

The analysis of the results of this *a priori* study shows that in all probability equipping vehicles with an anti-locking system provides advantages in safety equivalent to their cost and perhaps greater, especially for high-range vehicles.

The analytical method which we employed is concrete, but calls for the subjectivity of an expert. It may underestimate or overestimate the increase in safety provided by an anti-locking system, and it uses a rather arbitrary coefficient of maneuver adaptation. This is an initial approach enabling the question to be treated roughly and to determine that it is possible to undertake more costly studies ensuring more detailed results, taking into account actual behavioral adaptation and possible increase in risks taken by the drivers. To achieve this, an experiment must be carried out in which the safety of a group of vehicles equipped with such systems and of a control group can be compared. Such an experiment has been decided upon by the French Government.

The polling will have to take special account of accident distribution according to geographic location and seasonal periods.

The choice of a group of vehicles equipped with an anti-locking system is an important one, since the proportion of avoidable accidents varies from one to two between low-range vehicles and for high-range, front-wheel drive vehicles (from 16 to 29% if the drivers maintain calm enough to enable them to turn the wheel correctly).

Table I

Distribution, for Different Types of Maneuvers Performed, of Accidents Where the Anti-locking System Might Have Been Useful, as a Function of the Coefficients of Subjective Probability of Suppression of the Accident

Types of maneuver	Coefficients of subjective probability of suppression				
	0.1 to 0.3	0.4 to 0.6	0.7 to 1	Total	
Correct	17	13	32	62	16%
Incorrect	5	24	50	79	20%
No maneuver performed	50	71	74	195	50%
No manuever necessary	7	14	23	44	11%
Undetermined	3	5	3	11	3%
Total	82	127	182	391	100%

Table II
Distribution of Accidents by Types According to Whether the Anti-Locking System Might Have Been Useful or Not

Type of accident*	Accident where anti locking system might have been useful	Accident where anti locking system would have been useless	Non-conclusive studies	Total
1st	26 20.1% (6.6%)	70 54.3%	33 25.6%	129 100% (11.5%)
2nd	165 29.3% (42.2%)	390 69.1%	9 1.6%	564 100% (50.3%)
3rd	107 48% (27.4%)	104 46.7%	12 5.3%	223 100% (19.9)
4th	20 43.5% (5.2%)	24 52.2%	2 4.3%	46 100% (4.1%)
5th	40 55.5% (10.3%)	24 33.3%	8 11.2%	72 100% (6.4%)
6th	28 35% (7.1%)	42 52.5%	10 12.5%	80 100% (7.1%)
Others	5 71.4% (1.2%)	1 14.3%	1 14.3%	7 100% (0.7%)
Total	391 34.9% (100%)	655 58.4%	75 6.7%	1121 100% (100%)

*Definition of the types of accidents.

1 – Accident involving a single L.V. (loss of control)
2 – Accident involving an L.V. and at least one motorcyclist, cyclist or pedestrian
3 – Accident involving at least two vehicles (L.V. or heavy vehicle – H.V.) at an intersection (without overtaking)
4 – Accident involving at least two vehicles (L.V. or H.V.) following passing
5 – Accident involving a vehicle stopped on the roadway (H.V. or L.V.), or two vehicles driving in the same direction, without involving passing and not at an intersection (reduction in speed)
6 – Accident involving at least two vehicles (L.V. or H.V.) driving in opposite directions (without involving passing and not at an intersection)
7 – Others

Table III
Distribution of Avoidable Accidents by Type of Accident*

Type of accident*	Avoidable accidents	Remaining accidents	Total accidents
1	17 17.7% (7.2%)	79 82.3%	96 100% (9.2%)
2	103 18.6% (43.8%)	452 81.4%	555 100% (53.1%)
3	52 24.5% (22.1%)	160 75.5%	212 100% (20.3%)
4	15 35% (6.4%)	28 65%	43 100% (4.1%)
5	27 42% (11.5%)	37 58%	64 100% (6.1%)
6	19 27.1% (8.1%)	51 72.9%	70 100% (6.7%)
Others	2 33.3% (0.9%)	4 66.7%	6 100% (0.5%)
Total	235 22.5% (100%)	655 77.5%	1046 100% (100%)

N.B. *These results are deduced from Table II by application of the coefficients of subjective probability of suppression and considering that all preventive maneuvers performed by the driver are adapted to the circumstances.

Definition of the types of accidents is given in Table II

Table IV
Percentages of Avoided Accidents by Type as a Function of the Hypotheses of Adaptation of Maneuvers

Type of accident	Hypotheses of adaptation of maneuvers	
	Low (0.3)	High (0.6)
1	5.3%	10.6%
2	5.6%	11.2%
3	7.3%	14.6%
4	10.5%	21%
5	12.6%	25.2%
6	8.1%	16.2%
Total	6.7%	13.4%

Definition of the types of accidents is given in Table II

Table V
Distribution of Accidents as a Function of the Condition of the Road Surface According to Whether the Anti-Locking System Might Have Been Useful or Not

Condition of the roadway	Accidents where anti locking system might have been useful	Accidents where anti locking system would have been useless	Non-conclusive studies	Total
Dry	254 31.2% (64.9%)	520 64%	39 4.8%	813 100% (72.5%)
Wet	117 44.2% (29.9%)	119 44.9%	29 10.9%	265 100% (23.6%)
Others	20 46.5% (5.2%)	16 37.2%	7 16.3%	43 100% (3.9%)
Total	391 34.9% (100%)	655 58.4%	75 6.7%	1121 100% (100%)

Table VI
Distribution of Avoidable Accidents* as a Function of the Condition of the Roadway Surface

Condition of the roadway	Avoidable accidents	Remaining accidents	Total
Dry	160 20.6% (68.1%)	614 79.4%	774 100% (74%)
Wet, snowy, icy	75 27.6% (31.9%)	197 72.4%	272 100% (26%)
Total	235 22.5% (100%)	811 77.5%	1046 100% (100%)

N.B. *These results are deduced from Table V by application of the coefficients of subjective probability of suppression and considering that all maneuvers performed by the driver are adapted to the circumstances.

Table VII
Distribution of Accidents as a Function of the Condition of the Roadway and of the Coefficients of Subjective Probability of Suppression

Condition of the roadway	Coefficient of subjective probability			
	0.1 to 0.3	0.4 to 0.6	0.7 to 1	Total
Dry	43 16.9%	82 32.1%	129 51%	254 100%
Wet, snowy, icy	39 28.8%	45 33%	53 38.2%	137 100%
Total	82 21%	127 32.5%	182 46.5%	391 100%

Table VIII
Percentages of Avoided Accidents as a Function of the Condition of the Roadway Surface

Condition of the roadway	Hypotheses of adaptation of maneuvers	
	Low (0.3)	High (0.6)
Dry	6.2%	12.4%
Wet	8%	16%
Others	10%	20%
Total	6.7%	13.4%

Table IX
Distribution of Avoidable Accidents, for the Different Categories of Vehicles, by Equipping Them With an Anti-Locking System

Type of vehicle	Avoidable accidents	Accidents in which this type of vehicle is involved	
		No. of accidents	Proportion in relation to the total number of accidents
Vehicles no longer being manufactured (as of 1971)	38 (15.9%)	239 (100%)	22.8%
Low range vehicles	66 (16.1%)	410 (100%)	39.2%
Middle range vehicles, front wheel drive	26 (25%)	104 (100%)	10%
High range vehicles, front wheel drive	30 (28.8%)	104 (100%)	10%
High range vehicles, traditional drive	45 (22.7%)	198 (100%)	18.9%
Other French vehicles	6 (10.5%)	57 (100%)	5.5%
Foreign cars	24 (12.1%)	197 (100%)	18.9%
High range vehicles	75 (25.6%)	293 (100%)	28%
Middle and high range front wheel drive vehicles	56 (27.5%)	204 (100%)	19.5%

APPENDIX I

DISTRIBUTION OF THE 2,000 ACCIDENTS TO BE STUDIED IN EACH CENTER

The total number of accidents is distributed between the Surete Nationale and the Gendarmerie Nationale according to the ratio "France."

Two ratios are defined per department and compared to the national values.

$$R_1 = \frac{\text{accidents at an intersection}}{\text{total accidents}} \qquad R_2 = \frac{\text{killed + seriously injured}}{\text{total accidents}}$$

We are seeking by successive approximations the optimal solution which minimizes C in the following system

$$\sum_{i_1=1}^{4} R_{1.i_1} X_{i_1} = \bar{R}_{1.1} \sum_{i_1=1}^{4} X_{i_1} + C_1$$

$$\sum_{i_1=1}^{4} R_{2.i_1} X_{i_1} = \bar{R}_{2.1} \sum_{i_1=1}^{4} X_{i_1} + C_2$$

$$\sum_{i_2=1}^{4} R_{1.i_2} Y_{i_2} = \bar{R}_{1.2} \sum_{i_2=1}^{4} Y_{i_2} + C_3$$

$$\sum_{i_2=1}^{4} R_{2.i_2} Y_{i_2} = \bar{R}_{2.2} \sum_{i_2=1}^{4} Y_{i_2} + C_4$$

$$\text{with} \sum_{i_1=1}^{4} X_{i_1} = X = 1120 \text{ and } \sum_{i_2=1}^{4} Y_{i_2} = Y = 880$$

$R_{1.i_1}$ Value of ratio R_1 for the Surete Nationale (1) in department i_1
$\bar{R}_{1.1}$ Value of ratio R_1 for the Surete Nationale and for France
$R_{2.i_1}$ Value of ratio R_2 for the Surete Nationale in department i_1
$\bar{R}_{2.1}$ Value of ratio R_2 for the Surete Nationale and for France
$R_{1.i_2}$ Value of ratio R_1 for the Gendarmerie Nationale (2) in department i_2
$\bar{R}_{1.2}$ Value of ratio R_1 for the Gendarmerie Nationale and for France
$R_{2.i_2}$ Value of ratio R_2 for the Gendarmerie Nationale in department i_2
i_1 takes on the values 1, 2, 3, 4 in the optimal solution
i_2 takes on the values 1, 2, 3, 4 in the optimal solution
$X_1 . X_2 . X_3 . X_4$ size of the samples with respect to the Surete Nationale and to each department
$Y_1 . Y_2 . Y_3 . Y_4$ size of the samples with respect to the Gendarmerie Nationale and to each department

Selected distribution

Surete Nationale		Gendarmerie Nationale	
Val de Marne	400 accidents	Haute-Loire	100 accidents
Meurthe-et-Moselle	100 accidents	Cote-d'Or	225 accidents
Nord	400 accidents	Ille-et-Vilaine	200 accidents
Rhone	220 accidents	Haute Garonne	355 accidents

APPENDIX 2

POLICE ACCIDENT REPORT
GENDAMERIE NATIONALE

DOCUMENT #1, PAGE #1

Group 1

Company Squad

Case
S... T...
Report # 1970
Unit

Today,
We

having seen the articles of the penal procedure code, report the following operations which we have carried out, acting in uniform and in conformance with the orders of our chiefs.

	Date (in numbers)	Time	Time and source of the alert
of the accident	8-17-70	6:30 PM	6:40 PM by the fire brigade of
of the arrival on the scene	8-17-70	7:00 PM	

1. **Measures Taken Before Visiting Scene of Accident**

2. **Locale of Accident**
National Route #163 RENNES-CHATEAUBRIANT countryside

3. **Nature of Accident - Circumstances - Results - Measures Taken**
(In all cases of death or injury, indicate numbers and offices for social security or other organizations)

Nature of Accident:	Collision of two automobiles.
Circumstances:	A Car ... (A) driven by its owner Mr. S... coming from the direction of RENNES and heading toward CHATEAUBRIANT, swerved to the left in a a curve and collided with another car ... (B) driven by its owner Mr. T... travelling in the opposite direction.
Results:	One seriously injured: T..., driver of car B. Taken to the Clinique de la Sagesse at RENNES by the firemen of that city. Fractured pelvis. One slightly injured: S..., driver of car A. Received medical care at the University Hospital Center at RENNES, where he went of his own volition. Injury on the chin and thoracic contusions.
Measures Taken:	Traffic was kept flowing and directed in a single lane while the police investigation was being carried out. Photographs of the scene of the accident were taken. Both vehicles were removed by the garageman.

COPY OF MEDICAL CERTIFICATE
(to be used only if needed)

Dr. RENNES, 8-17-70

I, the undersigned, certify that I administered medical aid to Mr. T..., victim of an accident on August 17, 1970.

He showed evidence of a pelvic fracture with smashing of the cotyloid cavity and intra-pelvic dislocation of the left femur head, face wounds - multiple contusions.

This results in a 120-day period of work disability, barring complications.

Signed: illegible

Mr. S... having suffered no work disability, did not furnish a medical certificate.

4 Police Investigation: Scene Of The Accident

Atmospheric conditions	Clear and sunny weather, the sun being in the West, neither driver was blinded by it.
Route Name & No	NR #163, categorized for heavy traffic
Light	Daylight
Top View	Curve
Longitudinal	Flat road
Sectional	Slight banking
Road Condition	Good
Type of Surfacing	Asphalt
Condition of the Surface	Normal dry
Traffic Current	Traffic in both directions
Signalling	AB Panel (curves, the first of which is to the right, then to the left)
Location	J1 Beacon (curve-signaling beacon) Good visibility on both sides of the scene of the accident
Sides of the road	Grassy in part, with curb on the right side, in the direction Rennes-Chateaubriant Grassy shoulder on the opposite side
Shoulders (condition)	Good condition
Surrounding area	Fields
Barriers	None

Additional descriptions

It should be noted that National Highway 163 presents a very pronounced curve at the location of the accident. This curve is normally raised.

5 Police Investigation
Vehicles and Their Equipment

Vehicles' Usual Identification Letter (A, B)	Mr. S	Mr. T
Make and type	A	B
License No.		
For 2-wheelers motor and frame No.		
Driving side	Left	Left
Dimensions		
Gauge and wheelbase		
Lighting and signalling	Unverifiable because of the condition of the vehicle	Unverifiable because of the condition of the vehicle
Tires	Excellent condition for 3; left rear 70% worn, not blown	Good condition None blown
Speedometer (number indicated)	0	0
Kilometric indicator (number indicated)	78,567	67,194
Gear level	Neutral	3rd
Directionals	Unverifiable	Unverifiable
Brakes	Unverifiable	Unverifiable
Horn	No longer working	No longer working
Number of occupants (including driver)	One	One
Loading	Empty	Empty
Other equipment	No	No

Apparent damages, tracks, debris, general apparent condition of upkeep and maintenance, etc

	Entire front end smashed in. Some means of raising the vehicle needed in order to move it.	Entire front end smashed in; some means of raising the vehicle needed in order to move it.
	Seemed well cared for.	Seemed well cared for.
	Skid marks quite visible over a distance of 30 meters.	No marks.

6 Marks of Undetermined Origin and Other Findings

7 Administrative Information

DRIVER/USER

Name	Mr. S. . .	Mr. T. . .
First name		
Sex/Date of birth	M / 19 Sept. 1924	M / 18 May 1933
Place of birth		
Profession	Sales representative	Customs official
Nationality	French	French
Address		

DRIVERS' LICENSE

Number/Category(ies)	/Cat. B	/Cat. B
Date of issuance	1947	1959
Department		
Validity	Permanent	Permanent

OWNER

Name/Age	Same as driver	Same as driver
First name		
Address		

VEHICLE

Type/Make	Private automobile	Private automobile
Type/Series or frame no.		
2-wheel – motor no.		
Power/Weight/Capacity Seated Places	12 / 5	6 / 4/5
Date first registered	1967	1967
License number		
Date registration	1970	1967
Department	Ille-et-Vilame	Ille-et-Vilame
Inspection		

INSURANCE

Company headquarters		
Address of central office		
Policy number		
Risks covered	Contingent third-party liability	Third-party collision
Validity (from ... to ...)		

MISCELLANEOUS

Fiscal stamp	Valid	Valid
Papers		
Work legislation		

SKETCH OF THE CONDITION OF THE SCENE OF THE ACCIDENT

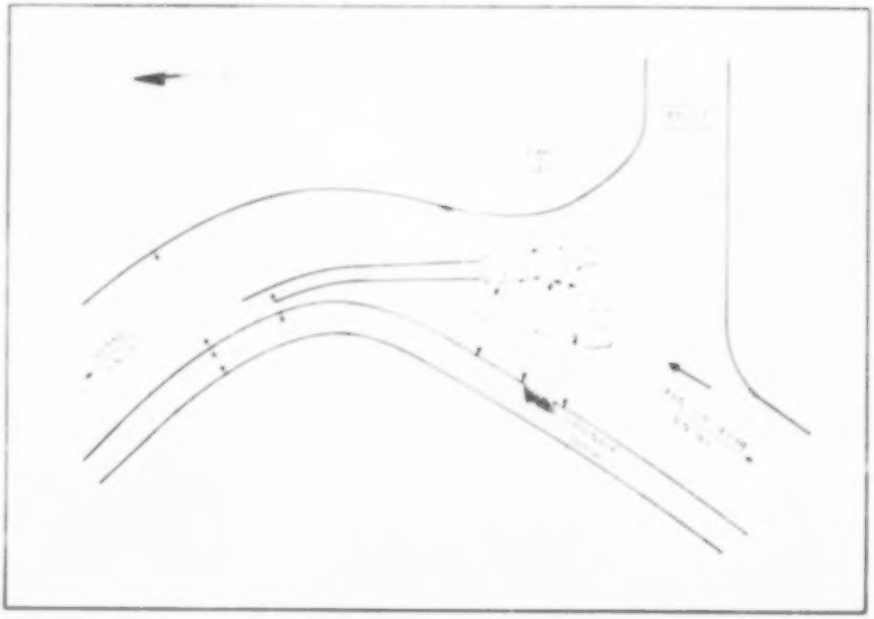
When using color in the execution of the drawing, red should only be used to designate the point of impact.

Base measures, taken from fixed points

from	to fixed point =	Distance in meters
left rear wheel axis vehicle **A**	1	7 m 10 cm
left front wheel axis vehicle **A**	1	6 m 25 cm
left front angle, vehicle **B**	1	7 m 75 cm

Legend Scale

Symbol	Designation
A	vehicle **Mr. S.**
B	vehicle **Mr. L.**
A10	skid marks from **A** over a distance of 30 meters
PF1	commercial sign **(ANTARGAZ)**



GENDARMERIE NATIONALE Case S... I

Group **REPORT ON THE HEARING** Report = 1970
Unit

Company Squad

PARTY HEARD: S
References

Today EIGHTEEN August 1970 at 10:30 AM
We the undersigned

Having seen articles 20 and 75 of the penal procedure code, report the following procedures which we have carried out, acting in uniform and in conformance with the orders of our chiefs, heard:
S..., sales representative, born September 19, 1924, declare:
"Yesterday, Monday, August 17, 1970, I was driving on National Highway #163, coming from **RENNES** and heading toward **VERN-SUR-SEICHE**. I was driving my personal auto, a ... (make) ... It was about 9:30 PM when, several hundred meters before arriving at the little town of **VERN-SUR-SEICHE** and after having, the best I could, negotiated the numerous and difficult turns in this road, I was surprised by a last, extremely bad curve. My car, which was travelling at approximately 90 km/h, was suddenly displaced and came face to face with another vehicle going in the direction of **RENNES**. I was unable, in spite of a most vigorous braking, to avoid the impact. I attribute this accident solely to the bad condition of the unmarked curve, and could not help noting that the numerous citizens who rushed to the scene of the accident were unanimous in stating that many drivers have often lost control of their vehicle at this particular place. I believe that my driving ability, given my job as a travelling representative, is sufficient enough for me to be able to affirm that such a curve is extremely dangerous and that for the accident in question it was the sole cause of the accident. I went on my own to the Hospital Center of **RENNES** where a complete and negative X-ray enabled me to go home at the end of the evening. As of this date no medical certificate has been delivered to me, under the condition, however, that unforeseen complications do not arise."

... August 18, 1970 at 10:30 AM ...

I have read the above declaration and find it to be accurate. I have nothing to change, add or omit thereto. (Signed the declaration book).

GENDARMERIE NATIONALE Case S L

Group **REPORT ON THE HEARING** Report # 1970

Unit

Company Squad

PARTY HEARD L

References

Today, August 22, 1970 at 10:10 AM

We the undersigned

Having seen articles 20 and 75 of the penal procedure code, report the following procedures which we have carried out, acting in uniform and in conformance with the orders of our chiefs, heard

L , customs officer, born May 18, 1933, declare

"On August 17, 1970, I was driving on National Highway #163 coming from the direction of **CHATEAUGIRON** and heading toward **RENNES**. I was alone in the vehicle. I was in third gear since I had just passed through the small town of **VERN-SUR-SEICHE**. Upon leaving this locality I saw, in a rather pronounced curve, a (make of car) which swerved to the left with respect to its driving direction. At first I thought that the driver would be able to regain control, but no doubt because of his speed or whatever, the vehicle smashed right into me. The impact was extremely violent. When I realized that nothing could be done to avoid the accident, I gripped the wheel. In this accident I received a fracture of the pelvis and I am suffering from contusions all over my body. I shall have a medical certificate delivered to you. On my part, I attribute all responsibility for this accident to the other driver. I reserve all my rights in this matter and plan to turn it over to a lawyer."

August 22, 1970 at 10:30 AM

I have read the above declaration and find it to be accurate. I have nothing to change, add or omit therein. (signed the declaration book)

APPENDIX 3

Some cases illustrating the a priori study method of the accident files.

1) The coefficient of subjective probability of suppression falls between 0.8 and 1.
The most typical situations found will be shown in the following examples.

- **Braking and Swerving**

Circumstances of the accident: An L.V. was travelling in a car lane. The traffic slowed down. The driver applied the brakes "energetically," the vehicle swerved to the left and hit a vehicle approaching from the opposide direction.

Use of anti-locking system: The source of the swerve was a locking of the wheels on a wet road surface. An anti-locking system would have enabled the driver to avoid the accident; the preventive maneuver would have not been needed.

- **Tangential Missing of the Curve**

Circumstances of the accident: The accident report presented in Appendix 2 is an illustration of this case.

Use of an anti-locking system: Tangential missing of the curve followed locking of the wheels. An anti-locking system would have enabled the driver to brake so as to reduce the speed of entry into the curve and thus negotiate it properly.

- **Reduction in Speed**

Circumstances of the accident: An L.V., during a slow down in traffic, ran into the rear of the vehicle in front of it. The skid marks were rectilineal and measured 31 meters.

Use of an anti-locking system: The preventive maneuver was possible to the left, without wheel locking. The time between the start and end of braking permitted the easy execution of this maneuver.

- **L.V. Hitting a Pedestrian or a Motorcyclist**

Circumstances of the accident: An L.V. hit a pedestrian who was crossing the road. The pedestrian was moving from left to right in relation to the direction in which the vehicle was travelling. The L.V.'s skid marks measured 20 meters. They began 0.60 meters from the right side and ended outside of the roadway on the side. The impact took place at the end of braking.

Use of the anti-locking system: The maneuver attempted was not adapted to the circumstances. The number of accidents with displacement of the vehicle in the direction of the displacement of the pedestrian, the cyclist or the motor-cyclist is considerable.

To avoid such accidents, it is necessary to:

equip the vehicle with an anti-locking system in order to make preventive maneuvers possible
instruct the drivers so that their preventive maneuvers are adapted to the circumstances.

- **Intersection**

In this type of accident, the maneuver was adapted to the circumstances and made possible avoidance of the accident. Wheel-locking was responsible for the failure of this maneuver.

Circumstances of the accident: To avoid a vehicle approaching his right, the driver braked, locked his wheels for about 40 meters and turned to the left. The vehicle turned over during this maneuver. The preventive attempt to the left was necessary because the vehicle refusing the right-of-way "jammed" on the right side of the roadway.

Use of antilocking system: An antilocking system would have enabled the driver to carry out the preventive maneuver.

2) The coefficient of subjective probability of suppression is equal to 0.5.

Circumstances of the accident: An E.V., which had reduced its speed in order to turn into a local road, was hit at the rear by a vehicle travelling in the same direction. Skid marks measured 28 meters and began 6 meters before the point of impact. The driver signaled the presence of a heavy truck travelling in the opposite direction.

Use of an antilocking system: The preventive maneuver made possible by the antilocking system was quite difficult to carry out. The point of impact was near the start of the skid marks, and the position of the heavy truck was not clearly indicated.

3) The coefficient of subjective probability of suppression is between 0.3 and 0.1.

Circumstances of the accident: Following a sudden reduction in speed in the left lane of the highway, a four-car pileup occurred. The skid marks of the last car measured 15 meters. The traffic was in two lanes. A safety guide was implanted to the left of the direction of the traffic.

Use of antilocking system: The preventive maneuver seemed extremely difficult to accomplish.

APPENDIX 4

Calculation of the mathematical expectancy of the actualized cost for the collectivity of accidents involving passenger vehicles.

In 1970, 236,109 bodily accidents occurred, for which there were 15,087 killed and 329,667 injured.

Accidents involving at least one passenger vehicle or one small truck accounted for approximately 84% of the accidents, 85% of those killed and 89% of those injured, or:

198,159 accidents
12,871 killed
293,387 injured

The cost in 1970 for corporal accidents involving a passenger vehicle or a small truck was 320,000 x 12,871 + 14,000 x 293,387 + 5,600 x 198,159 = 9,336 x 10^6.

The cost of bodily accidents per passenger vehicle or small truck in circulation in 1970 was 9,336/14 = 667 F. Bodily accidents represent approximately 7% of the total accidents.

Material accidents can be estimated at 2.6 millions, for an average tax-free cost of 700 F per accident.

The cost in 1970 of material accidents involving a passenger vehicle or a small truck was 2,600,000 x 700 = 1,820 x 10^6 F.

The cost of material accidents per passenger vehicle or small truck in circulation in 1970 was:

1,820/14 = 130 F

Thus, the cost of accidents per passenger vehicle or small truck in circulation in 1970 was:

667 + 130 = 797 F

This is an estimate of the mathematical expectancy of the cost of annual accidents for a passenger vehicle or a small truck, assuming that the probability of accident is the same for all vehicles. If it remains constant like this for the ten years of its average life (when the vehicle ages, the increase in the rate of accidents per kilometer driven is compensated by the decrease in the annual average distance driven), the mathematical expectancy of the cost of accidents of the vehicle involved, actualized at the rate of 10% is:

797 x 6.14 = 4,900 F

THE FRENCH TECHNICAL PRESENTATION

SECTION TWO PART 3

RESTATEMENT OF PEUGEOT-RENAULT ASSOCIATION'S PROGRAM

GEORGES BOSCHETTI, *Director*
Paris Study Center
Peugeot Renault Association

In order to make Peugeot-Renault Association's research works quite clear and show the continuity of its program, I shall remind you of the first steps.

At the Paris meeting in January 1971, we were somewhat surprised to see that the ESV proposals, which had just been stated, resulted in particularly heavy vehicles. We demonstrated that with respect to side impact there was also a problem of aggressiveness on account of the low thickness of the doors in relation to the far larger dimensions of vehicle deformable front and rear parts.

Because of this, the NHTSA conferred on us the title of pioneer, and we were very proud of it.

In Stuttgart, during the Second International Conference, we expressed our work program:

- Unrestrained research to experiment without being unnecessarily restricted by the specifications.
- To study closely the relations between cost and benefit by developing investigations on real accidents and by determining carefully the cost of the contemplated solution.
- To make an effort to apply the solutions to future models, with more integrated solutions as soon as possible.

We proclaimed the priority of ESSS work relative to ESV work, expressing the wish that the two methods be not opposed, but also that the second be not preferred at the expense of the first one.

In Washington, only eight months ago, we dealt with the most difficult problem of collisions between big and small cars, showing up the necessary reduction of big car front end stiffness and the increase of smaller car stiffness.

We expressed our opinion on several points, particularly in regard to the study of real collisions, through an increase in accident investigations and the study of aggressiveness, which has since been largely developed throughout the world.

Presently, we carry on the analysis while preparing at the same time the synthesis, but this synthesis is not always very easy. We think that it is more intricate, more difficult to specify than is the drawing and materialization of experimental vehicles. One of the greatest difficulties in determining specifications is estimating real collision severity for the occupants, because the levels of the useful performances will have to be determined.

Mr Prost-Dame, Head of Vehicle Body Study and member of Industrial Applications of Renault will submit the first conclusions of such a difficult study. The tests and the studies, conducted for this presentation were parcelled out between both members of the Association.

The "Regie Renault" carried on the work of analysis. Mr Ventre, Head of Draft and Safety Test Department, will present the following works on the compatibility between vehicles of different types, passing from the notion of aggressiveness to the notion of compatibility, which is basic if we are to avoid the problem of increased safety in one vehicle at the expense of safety in other vehicles.

Peugeot Automobiles examined the problem of synthetic ESV vehicles. Mr Hamon, Head of ESV Draft, will demonstrate how it is possible even from the partial studies and the research tests, to transpose the ideas on the ESV synthesis with reasonable accuracy.

He will discuss the case of the smallest and largest cars and will principally describe the medium program of the 1,000 kg vehicle from a basic mechanical system of Peugeot 304 type with a transverse engine.

The performance tests chosen in this study are only hypothetical and do not correspond to any official specifications.

Mr Hamon will demonstrate that the study of a complete synthetic vehicle, which is by definition different from the real prototypes which remain secret for obvious commercial reasons, creates as many new problems, by going away from future reality, as it helps to resolve. As a matter of fact, the ESV can only be studied as research work. It is to be presented to the government, the press and the public, and, out of necessity, must therefore have different shapes and dimensions from real prototypes.

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These differences separate the ESV research work from the real industrializable synthesis, even though this work can be carried out in a far more useful way on subsystems which respect, locally, the same solutions in the same dimensions as those contemplated for future production. Moreover, these big subsystems allow an increase in the number of solutions for a similar test. From a technical point of view they allow the best orientations.

On the contrary, the complete ESV, in which it is possible to sit and drive, allows a realistic approach to the problems and their contradictory aspects, not only for the manufacturers but, also, for the government, the press and the public. It reveals the total result on the weight, volume and price of vehicles as well as the corresponding increase of the powers necessary to ensure the same performances. It also allows a better orientation of decisions from the economic and social standpoints.

In which direction have we to carry on? This is a question which the Peugeot-Renault Association asks itself.

It will not conclude and will, on the contrary, leave the question very open.

- The best syntheses as regards cost-benefit ratio are the syntheses on big subsystems. It matters little whether they are called big ESSS or partial ESV.
- The complete synthetic vehicle presents unquestionable interest: it is spectacular and it has been the noteworthy advertising agent for the research program on safety, and possibly for the manufacturers which consented to it before making the analysis by subsystems.

It has been an excellent propulsive in the world. It is what we may call "a producing myth". But we do not forget that most useful will probably remain the synthetic prototype.

THE FRENCH TECHNICAL PRESENTATION

SECTION **TWO** PART **3**

ESV SYNTHETIC DRAFT

JEAN HAMON, *Head*
Structure Aerodynamics Department
"Centre d'Etudes de Paris"
Peugeot

SUMMARY

Most of the solutions developed on experimental safety sub-systems (ESSS) are concentrated in the Safety Car studied by Peugeot. The aim is to find a technical compatibility between these sub-systems which is bound to the economic imperatives of a cost-efficiency ratio adapted to the European Market.

ESSS STUDY BALANCE

These studies have been performed separately but synthesized with respect to structure and occupant protection.

Structure

The criteria retained were only bound to the deformation; the injury criteria were not taken into account.

For frontal impact testing, we adopted a pole impact, an inclined wall impact and naturally a fixed barrier direct impact with severity levels up to 80 km/h.

ESSS lateral impact has been studied on the basis of vehicle collisions to take into account both stiffness and aggressiveness.

As regards rear-end impact, the study was carried out with a double aim: to avoid fuel spillage and to reduce passenger compartment internal distortions.

Occupant Protection

Our action was performed on two levels:

- Active and passive protection systems for severe impacts (air bags, unfolding partition walls, safety seats, etc...).
- Passive protection in urban collisions with deformable steering wheel, windscreen, energy absorbing [illegible] boards, knee restraint systems, etc...).

The studies are going well and the solutions drawn from them may now be integrated in an ESV Synthetic Draft.

PEUGEOT SAFETY CAR

Performances reached in ESV specifications are only Peugeot hypotheses; they are not linked with any external specifications but they have been determined in view of the first results from inquiries on real accidents.

They concern more specifically:

Primary Safety

- Road comportment (road holding, braking, steering precision and softness).
- Direct and indirect vision.
- Lighting and signalling.
- Control disposition and accessibility.
- Level of sound and comfort.

Secondary Safety

- Protection of occupants during impact through compliance with injury criteria and after impact by providing for rapid removal from the crashed vehicle.
- Reduction of aggressiveness towards other vehicles and pedestrians.

To improve safety performances homogeneously, Peugeot applied the principles defined by ESSS to all models of its range, but only the draft relative to a middle car similar to the "304" is detailed. It is a four-seat sedan with four doors, front transverse engine and front wheel drive.

Technical knowledge acquired from ESSS allows us to study and define a synthetic vehicle with all the advantages bound to an ESV study to:

- Verify technical compatibility of partial solutions.
- Improve advanced techniques.
- Promote exchanges and diffusion of knowledge, this type of draft being non-confidential.

Yet, ESV presents obvious disadvantages:

- It necessitates additional means.
- There exists, inevitably, a difference between ESV and future models, and we wonder whether we have to carry out the synthetic ESV with the benefits drawn from knowledge acquired on ESSS, for the main problem is a realistic application to current and future production vehicles.

Our "Safety Vehicle Draft" results from all the partial studies that have been performed on ESSS for more than two years. The studies have allowed us to determine maximum performances and the corresponding technical solutions for each vehicle subsystem playing a basic part in "secondary safety," such as:

- Front-end structure for frontal impact and reduction of aggressiveness.
- Rear-end structure for rear impact and fire prevention.
- Restraint systems for severe impacts.
- Passive protection in urban collisions, etc.

These studies have been rather successful in most fields; they have been carried out on the basis of arbitrarily high technical specifications. Checking the efficiency of the corresponding solutions on a synthetic vehicle logically follows with two consequences:

a. *at the technical level:* investigation with regard to compatibility between the performances achieved for each ESSS and, in particular, determination of the best possible compromise between the stiffness of structures and the efficiency of restraint systems.

b. *at the economical level:* selecting solutions that lead to a better benefit-cost ratio. In this field we still lack clear enough data, and we are making the data more precise by means of different studies of real accidents carried out at the European level, particularly by the Peugeot-Renault Association and centralized at CCMC's level.

Contrary to ESSS, the draft studied by Peugeot is not to be considered as an absolute study whose objective is the highest possible level of technical performance. It is, rather, the prefiguration of a future vehicle which takes advantage of considerably improved safety features but remains reasonable in regard to its possible producibility and profitability on the European market.

PRESENT STATE OF STUDIES ON SAFETY SUB-SYSTEMS

Structure Study

Frontal impact sub-system

1. Specifications:

The front-end structure has been tested against three types of obstacles: pole, 30° wall and front wall collision in the following speed ranges: 50 km/h, 65 km/h and more than 65 km/h (Figure 1). We decided upon two levels of performances:

a. Keeping sufficient space for proper unfolding of restraint systems which is checked by measuring the percentage reduction of the front seat survival space and preventing possible openings of doors after impact.

b. As a safety coefficient, preventing collapse of the passenger compartment under the most severe impact conditions.

2. Structure comportment:

- In a direct frontal impact, the impact energy is absorbed by the buckling of the engine-supporting cradle and the front side rails and by deformation of the upper box-like parts of the outer wheelhouse and wheel compression on supporting beams housed in floor reinforcements.
- In a 30° barrier impact, the most acted upon side works substantially in the same way. However, the wheel supporting beam action becomes more important: at the end of impact it prohibits wheel penetration into the apron and facilitates vehicle stopping (Figure 2).

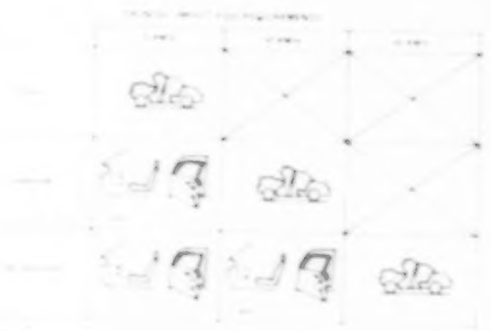

Figure 1

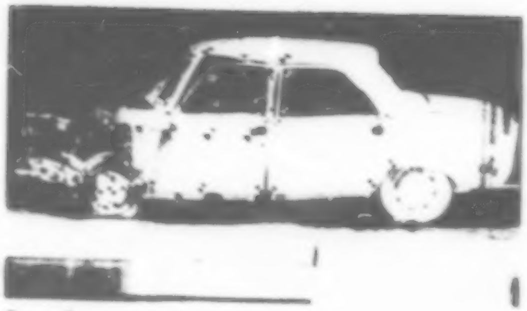

Figure 2

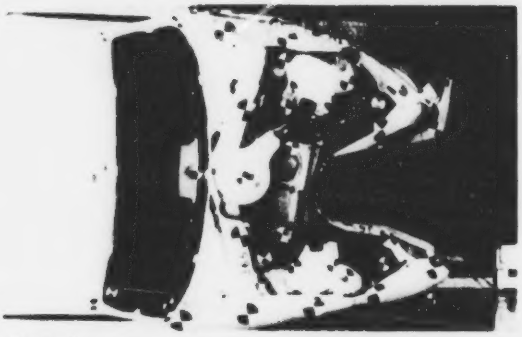

Figure 3

FRONTAL IMPACT - RESULTS STRUCTURE BEHAVIOR			
	50 KM/H	65 KM/H	>65 KM/H
POLE	NO DESTRUCTION		
30° BARRIER	9% < 30%	NO DESTRUCTION	
FRONT BARRIER	7% < 15%	6% < 30%	NO DESTRUCTION

Figure 4

- In a centered pole impact, the transversely-mounted engine keeps the two sides of the front block separated and enables bending of the side-mounted structure reinforcements. Moreover, it rests upon the rear cradle cross member and stops the pole from reaching the apron (Figure 3).

3. Test results:

The goals we decided upon have been reached:

- Survival space reductions are far below imposed limits (Figure 4):
  7% at 50 km/h against fixed barrier
  9% at 50 km/h against 30° wall
  only 6% at 65 km/h against front wall after improvement of structure.
- After the 50 km/h front wall tests, we modified the antilock door device to provide for its opening under conditions far more severe than those initially contemplated (Figure 5).
- No collapse of the passenger compartment occurred, even during an 80 km/h front wall test.

FRONTAL IMPACT – RESULTS NUMBER OF DOORS OPENING AFTER IMPACT			
	50 KM/H	65 KM/H	> 65 KM/H
POLE	4		
30° BARRIER	2	4	
FRONT BARRIER	2	4	2

Figure 5

Figures 2 and 3 show the ESSS appearance after an impact test at 65 km/h against a 30° wall and at 50 km/h against a pole.

But the results achieved comply with our aims only at the cost of a substantial increase in the stiffness of structure reinforcements and consequently of deceleration levels.

During a 61 km/h front barrier test the passenger compartment underwent a 50 g deceleration peak. At 80 km/h the peak exceeded 120 g; it reached 40 g for a 50 km/h impact.

As a result of the tests carried out with dummies, it was found that such an impact severity poses difficulties with occupant survival. This problem must be resolved on a synthetic vehicle.

Our ESSS tests also confirmed the well-known fact that for a given impact speed decelerations are lower with a 30° barrier impact or with a "pole" impact than with a direct barrier impact. But the facts prove that offset frontal collisions or narrow obstacle collisions (tree, pole...) are largely preponderant on a road. This configuration must be taken as a reference so that the best technical compromise can be reached; that is, to have the maximum structure stiffness allowable yet comply with injury criteria.

It is obvious, therefore, that in direct barrier impacts, which, fortunately, are very rare, these injury criteria will only be met at lower speeds because of the correlative deceleration rise.

Homogeneous specifications must therefore provide for offset frontal impacts or "pole" impacts at speeds higher than those specified for fixed barrier impact tests.

Side Impact Sub-system

This study was conducted on the basis of vehicle collisions, in order to take into account the stiffness of an impacted vehicle side and the aggressiveness of an impacting vehicle's front end.

This aggressiveness was reduced through reinforcement of the lower part of the front framework which pushes against the impacted vehicle floor in a high strength area.

During tests performed at 50 km/h with angles of incidence of 45° and 90°, we obtained a low intrusion into the body size (Figures 6 and 7) which was below the 100 mm limit fixed.

Comparisons were made on the comportment of dummies, both in current and reinforced vehicles. Results show that the occupant of a current vehicle seems to be substantially pulled at the pelvis and neck levels. In a reinforced car, occupant motion is limited.

The latter, however, impacts on the thigh side with a violence that can become great when the vehicle is reinforced.

Within those limits there exists an optimal reinforcement level depending upon the geometry of the vehicle, the stiffness of the inner walls and the impacting vehicle.

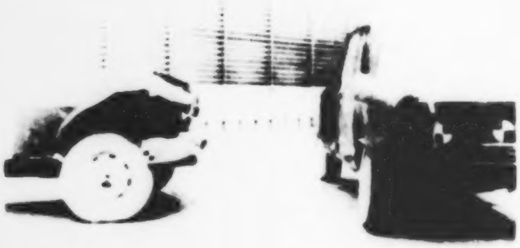

Figure 6

SIDE IMPACT
50 KM/H
TEST RESULTS

45°	**INTRUSION**	**60 MM < 100 MM**
90°	**INTRUSION**	**20 MM < 100 MM**

Figure 7

Pending the definition of aggressiveness criteria, we think that tests should be carried out with a couple of vehicles of the same type. The specification to be used in the absence of injury criteria adapted to side impact is a limitation of the inside door panels' intrusion.

Rear Impact Sub-system

As regards rear impacts, our primary objectives were to provide fire protection by avoiding fuel leakage and to obtain low deformation of the rear seat substructure, in any case a deformation of less than 10 cm.

At a speed of 50 km/h, the flat-faced moving barrier crushed the rear end of the vehicle to 85 cm (Figure 8) without causing fuel leakage or damage to the fuel system or tank. This was achieved by causing the floor and structural reinforcement to buckle and absorb energy without penetrating the passenger compartment or deforming the rear seat supporting structure or doors. The doors, incidentally, could be opened after the test (Figure 9).

During such an impact, dummies are not submitted to high deceleration forces, but, in the absence of head restraints, they suffer from severe neck hyper-extension.

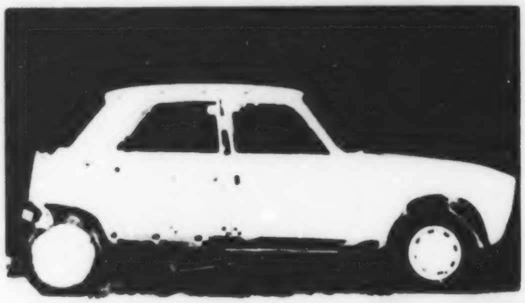

Figure 8

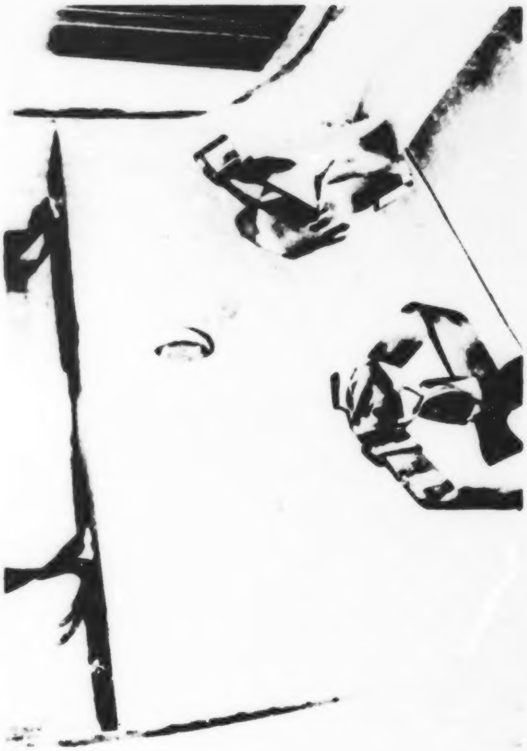

Figure 9

Occupant Protection

Our specifications concerning occupant protection are the currently used injury criteria:

head: Gadd and HIC indices below 1,000,

thorax: acceleration module no higher than 60 g during more that 3 ms and Gadd index lower than 1,000,

thigh-bones: compression force lower than 6,350 N.

Safety Seat

The safety seat makes it possible to provide for a very good pelvic restraint and a correct head path owing to the optimal arrangement of belts relative to the pelvis and thorax.

When distorting, the seat frame absorbs energy, which enables it to limit stresses applied to occupants.

During tests performed at 50 km/h, on a carriage and in a passenger compartment (Figure 10), with a decelerating device which reproduced deceleration encountered in vehicles, we obtained injury criteria meeting our specifications with a substantial safety margin (Figure 11).

Within the passenger compartment, the energy-absorbing knee restraint device holds the thigh-bones in and alters the dummy trajectory so that the head will not impact the windshield.

Figure 10

SAFETY SEAT INJURY CRITERIA	
HEAD:	SI 850/HIC 750
TORSO:	46 g/3ms. SI 400
FEMUR FORCE:	4000 N

Figure 11

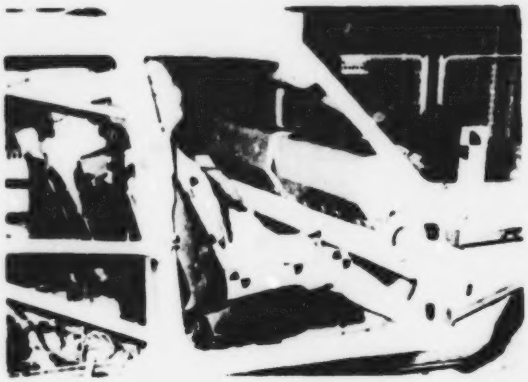

Figure 12

Passive Protection at Low Speed

Because belts are not frequently worn in urban centers, an occupant passive restraint must be provided at low speed impacts (32 km/h). In this sub-system, passive protection is provided by a windshield whose inner surface is designed to eliminate lacerations to the face, by an energy-dissipating dash board and by a knee restraint system (Figure 12).

In case of impact, the legs push against the restraint plate and couple the passenger with the vehicle during the last phase of front-end deformation. Then, the thorax and head rest upon the dash-board and windshield, the latter acting as a kind of net.

Deformation and energy absorption capacity of these three components prevents hyper-extension of the spinal column, and limits occupant rebound. Moreover, the windshield inner-face efficiently protects the head against any risk of intrusion or laceration.

With such a configuration, performances achieved are substantially compatible with human tolerances (Figure 13), and should provide substantial improvement in safety during urban collisions.

PASSIVE PROCTECTION
AT LOW SPEED
INJURY CRITERIA

HEAD	HIC 500
TORSO	38 g/3 ms. SI 500
FEMUR FORCE	4000 N

Figure 13

Peugeot Safety Vehicle Program
General Information

The solutions developed on ESSS, an example of which was described above, are to be applied to the safety vehicle draft program.

For this program, we paid particular attention to the following items:

Regarding primary safety

- Highway comportment (road holding, braking performance, accuracy and smoothness of steering);
- Direct and indirect visibility;
- Lighting and signalling;
- Lay-out and accessibility of controls;
- Sound level and comfort which have an influence upon the driver with respect to fatigue caused by nervous tension
- Aid to driving at high speeds.

Regarding secondary safety

- Occupant protection during impact through compliance with known human tolerance criteria and after impact by making sure that occupants are rapidly rescued from the smashed vehicle;
- Reduction of aggressiveness, both towards other vehicles and pedestrians.

Impact Performance Criteria

In *frontal impact collisions* (Figure 14), we retained only the most frequent highway accident configurations, namely offset frontal collisions and "pole" collisions. However, we maintained the conventional fixed barrier impact test.

For three types of impact at 50 km/h (fixed frontal barrier) and at 55 km/h (30° barrier and pole), we request that sufficient space be provided to allow for unfolding of the restraint means and that injury criteria be complied with.

Moreover, with a view to providing a safety factor for passenger compartment strength, we request that the latter not collapse during impacts at higher speed (about 65 km/h against fixed barrier).

Doors must not open during impact tests and at least one of them must be opened both from inside and outside following the collision.

Figure 15 shows that in lateral collisions the interaction between the *impacting* vehicle's front structure and the *impacted* vehicle's side structure plays a basic part. For this reason, as it was in the case for the ESSS, we rejected the moving barrier test, because it would be either too aggressive, if it had a front vehicle shape, or if flat insufficient to estimate the strength of the body side.

We adopted a *90°* lateral impact between two similar vehicles which insured that:

- the body side will be resistant enough;
- the aggressiveness will be reduced during lateral impact due to the fact that it has to act as a buffer against its own side.

In a rear impact, by a 1,800 kg moving barrier at 50 km/h (Figure 16), we should meet injury criteria and door requirements. Moreover, in connection with fire prevention, no fuel leakage will be allowed.

ESV REQUIREMENTS
FRONTAL IMPACT

	50 KM/H	55 KM/H	65 KM/H
POLE		INJURY CRITERIA	╳
30° BARRIER		INJURY CIRTERIA	╳
FRONT BARRIER	INJURY CRITERIA	╳	NO DESTRUCTION

Figure 14

ESV REQUIREMENTS
SIDE IMPACT
ESV/ESV COLLISION
V = 50 KM/H
INSIDE DOOR PANEL INTRUSION < 100 MM*

*TO BE REPLACED BY INJURY CRITERIA WHEN DEFINED

Figure 15

ESV REQUIREMENTS

REAR IMPACT
- MOVING BARRIER V = 50 KM/H
  INJURY CRITERIA
  NO FUEL LEAKAGE

ROLL-OVER
- STATIC ROOF STRENGTH
- DYNAMIC ROLL-OVER
  NO OCCUPANT EJECTION
  NO FUEL LEAKAGE

Figure 16

Rollover French statistical data show that 75% of fatal and severe injuries in rollover accidents result from the ejection of occupants. We think it is absolutely necessary to give full priority to this problem, particularly by preventing doors from opening during rollover.

It is also highly advisable that the roof be tested for minimum strength by a static compression test.

Urban collisions For collisions at very low speeds, a slight structural distortion will be allowed, but it is requested that following a 10 km/h impact against a wall either in forward or reverse, the vehicle be in a position to start again with all its safety components (brakes, suspension, steering) in working order. Lights and headlamps should not be damaged and should be re-aimed if necessary.

For more severe collisions (32 km/h) interior equipment will be required to provide passive protection whenever occupants are not fastened.

The above specification should strongly increase safety in urban accidents.

General Lay-out

With a view to developing homogeneous safety performances, the results achieved on ESSS are applied at various levels of the European range.

On top-range vehicles (Figure 17), that is, vehicles of 1,500 kg or more, it is necessary to counterbalance the undue influence of their mass towards small cars by reducing aggressiveness. This may be done by limiting frontal structure stiffness of the big vehicle by using the greater crushing distance available on those cars. We also consider the possibility of using hydraulic bumpers (Figures 18 and 19) mounted on bigger vehicles to obtain an artificial increase of deformation capacity with a view to dissipating impact energy under moderate loads acceptable for smaller cars.

Medium-range vehicles of the range (1,000 to 1,500 kg) (Figure 20), form the subject of the detailed draft described below.

Lastly, as regards low-range vehicles (Figure 21), we do our utmost to find out the best possible weight/performances ratio for the structure. On these models, the balance between restraint performance and structure stiffness is most difficult to achieve.

General Features

The following draft ranges substantially on the same level as the 304, whose basic lay-out is used for the ESV.

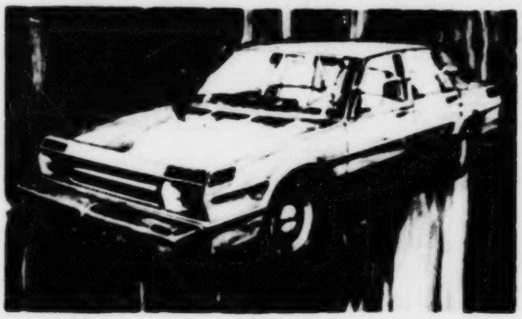

Figure 17

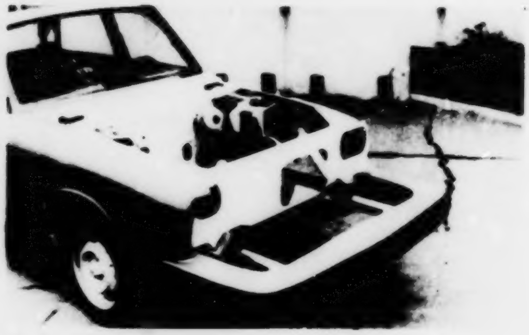

Figure 19

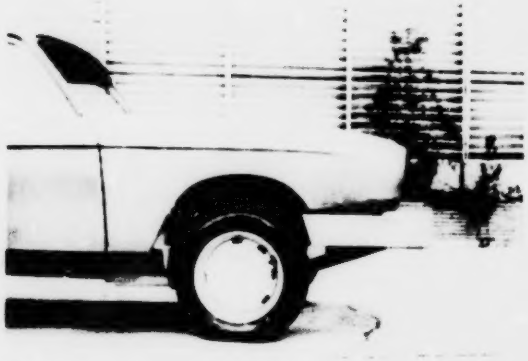

Figure 18

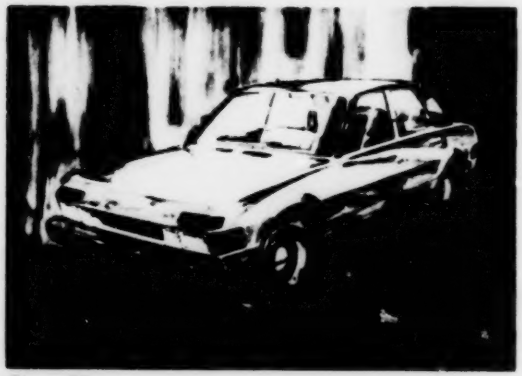

Figure 20

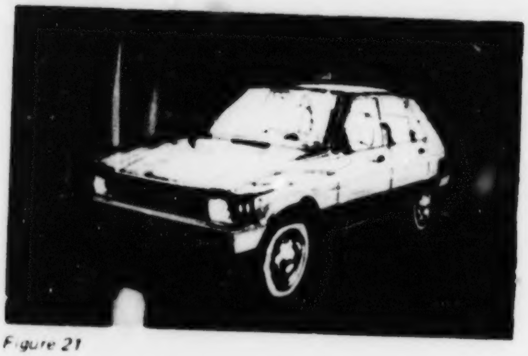

Figure 21

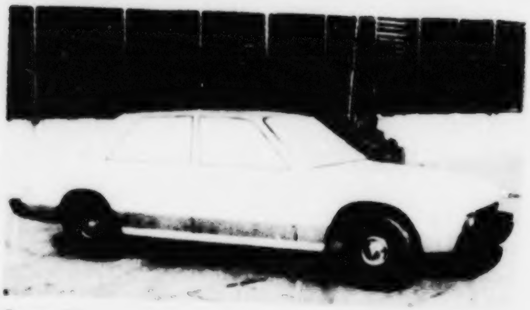

Figure 22

The ESV is a four-door sedan (Figure 22), with only four seats. It is a front-drive vehicle with a transversely mounted engine.

Its main data (Figure 23) are summarized in the table below.

Dimensions

Overall length	4.590 mm
Overall width	1.650 mm
Wheelbase	2.700 mm
Front wheel track	1.440 mm
Rear wheel track	1.400 mm

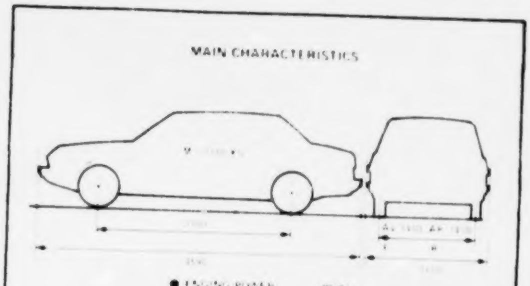

Figure 23

Weight	Unloaded weight	1.150 kg
	Curb weight	1.490 kg
Engine power		90 CV
Performances		160 km/h

We are now working to establish specifications relating to aerodynamic (C_x, C_y, C_z, . . .) performance and necessary road holding qualities:

- stability against cross-wind
- and, regarding the fastest models, avoiding load decrease for front and rear axles.

Structural design

Against frontal impacts (Figure 24) two series of strongly-built elements are respectively arranged on the upper part (box-type reinforcement of outer wheelhouse) and on the lower part of the front structure. With a view to reducing aggressiveness in case of side impact, the lower element projects substantially beyond the upper element (a special "anti-aggressiveness" cross-member enables us to push an impacted car by its floor rather than by its doors).

In the case of a frontal impact, stresses are supported at the passenger compartment level:

- in the upper part by longitudinal door reinforcements.
- in the lower part by beams under the front floor (Figure 25) which compress the floor in its own plane and transmit stresses to side rails in the lower part of the body.

Box-type elements mounted at the back of the wheels prevent their intrusion into the passenger compartment during offset impacts.

The front upper-frame cross member which is reinforced and firmly fixed on the upper box-type element of the outer wheelhouse enables bending of the latter during "pole" impacts.

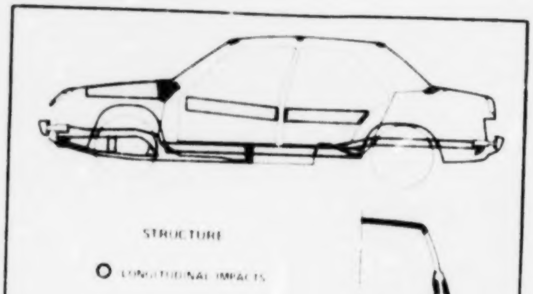

Figure 24

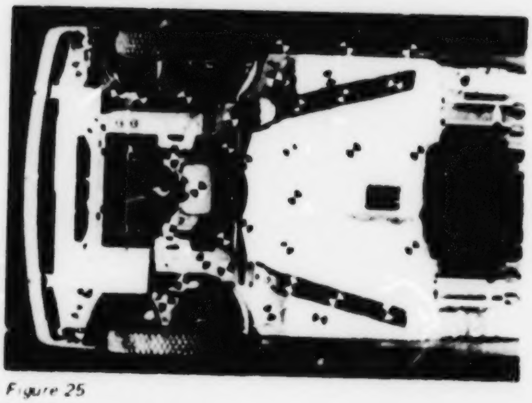

Figure 25

In a side impact, door panel intrusion is reduced by means of the same door reinforcements used in a frontal impact, with the exception that this time they act like a chain which is anchored between the front and rear ends of the passenger compartment.

On the other hand, we contemplate transverse box-type elements which work by compression to link both body sides together; this effect is completed by the seats whose substructure is reinforced to constitute a compression chain acting as a cross member between sides of the body.

In a rear impact, the trunk compartment floor is reinforced by two beams. Its buckling is controlled so that the floor crumples up under impact in order to prevent any substantial intrusion into the passenger compartment. The fuel tank is protected by two lateral, box-type, elements mounted on the rear seat substructures which transmit forces to the side rails on the lower part of the body. Various locations for the tank are satisfactory from the point of view of fire prevention:

- Between the rear suspension arms (protection is insured by very rigid arms).
- Under the rear seat supporting structures (protection is by a strongly reinforced body side structure).
- Behind the seat back plate (furthermost location from eventual impact point).

In case of roll-over. Transverse box-type elements and a roll-bar are provided between the B pillars to insure good passenger compartment performance.

Occupant Protection Equipments

a. *Safety Seat* (Figure 26). We had the choice of completely passive devices (air-bags, etc. . .) and the more classical active devices.

Passive restraint systems, though very efficient in the case of purely frontal impact, show substantially reduced performance in all other accident configurations. On the other hand, their cost is very high.

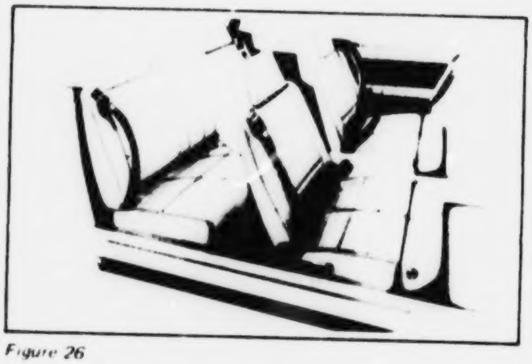

Figure 26

On these grounds, we resorted to the more classical restraint systems, namely safety belts equipped with a retractor integrated with the seat and used in conjunction with a knee restraint system (Figure 27).

The solution described above has the following advantages:

1. Easy to handle when compared with conventional 3-point belts, strap positioning remains correct whatever the occupant size may be.
2. Seat back structure is specially designed to accept limited deformations during impact; deformations induce a degradation of energy higher than with a classical belt retractor, which results in strongly improved performance.
3. Upper anchorage location of the diagonal strap on the back of the seat enables better control of trajectory and prevents projection of the head against windshield.
4. The cost of such a device, though higher than that of conventional belts currently mounted

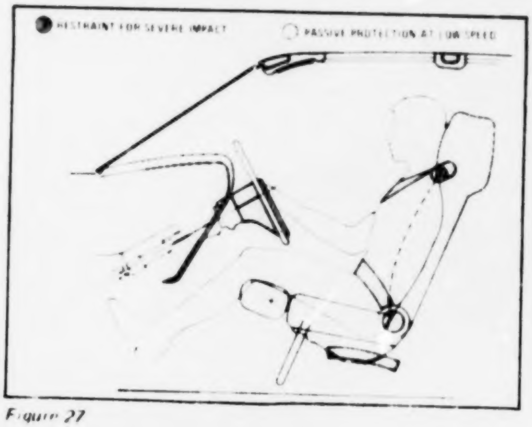

Figure 27

on our production vehicles, remains far below that of air-bags.

b. *Interior Equipment* (Figure 28). The various equipment provided within the passenger compartment is intended for various functions according to the type of impact.

Severe Impacts. the restraint system is associated with the knee restraint, which is distortable enough not to interfere with occupant thigh-bones, so long as it is not submitted to forces exceeding known tolerances, that is 6,350 Newtons.

As for the driver, the steering-wheel is equipped a shock-absorbing padding for the head in impacts.

Medium speed urban impacts (32 km/h). passive safety, regarding *non-restrained* occupants, is insured by:

- a distortable dashboard of convenient size whose role is to decelerate the thorax while dissipating energy;
- for the driver, a tulip-like steering-wheel frame distortable under slight stresses which has the same role as the dashboard;
- above-described knee restraint metal plate;
- utilization of Securiflex glass for the windshield. This is a laminated glass made of an external sheet of glass and an internal plastic sheet. This assembly eliminates any risk of perforation though resistant to substantial distortions.

The various structural elements projecting into passenger compartment, as well as head-rests and front seat rear panels, are covered with an energy-absorbing protecting material (only 5% of energy restituted).

c. *Steering Column* The steering column has two functions to protect the driver during second impact (Figure 29).

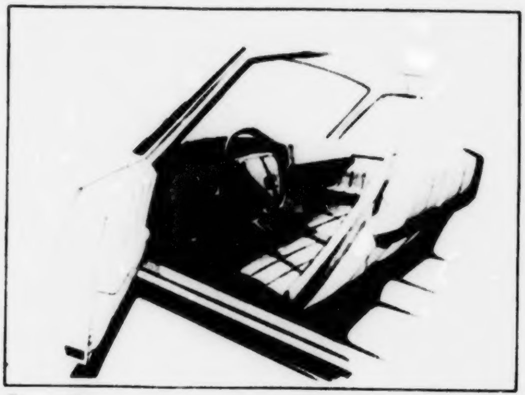

Figure 28

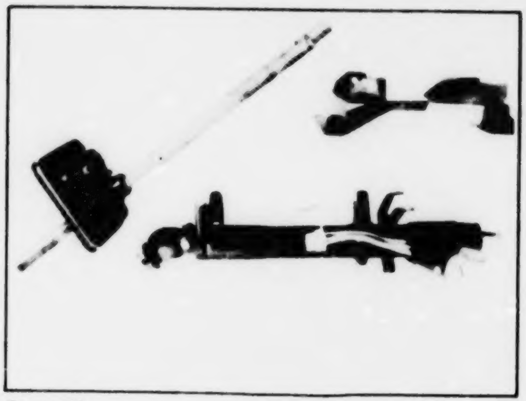

Figure 29

If the driver is fastened his thorax does not impact against the steering column and it is sufficient to prevent an eventual head impact. An unrestrained driver is provided for by a distortable column fastener which enables the column to slide forward under a 500 to 700 kg force which the thorax can withstand because of the large bearing surface of the steering wheel. The fastener is rigid in case of forces exerted in the reverse direction. The wheel (Figure 30) is made on a "steering" distortable structure; it also contributes to energy absorption.

To prevent rear displacement of steering column within the passenger compartment, a dual universal joint mechanical drive is used. The shaft between the joints is slidable so that the upper joint remains fixed regardless of the lower joint movements.

Primary Safety Elements

We shall not dwell upon technical details concerning mechanical components and the power unit.

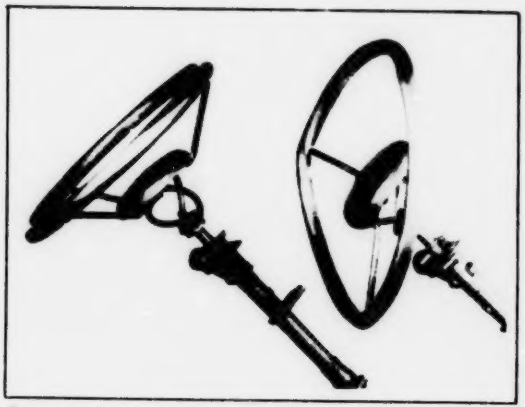

Figure 30

Among well-tried solutions, we selected those providing the best compromise between road holding qualities, driveability and braking performances. In this latter field we contemplate, among others, a dual braking system and an antilock device, independent on the front wheels and single on the rear wheels.

We also deal with the utmost care with lighting and signalling problems:

- at the front (Figure 31), the ESV is equipped with halogen headlamps including an automatic re-aiming device;
- side parking lamps with built-in reflectors are integrated into front and rear signalling boxes;
- at the rear (Figure 32) a composite signalling assembly including:
  - stop lamp
  - rear fog lamp
  - backing lamp
  - tail protection and parking lamp

The latter is a rear-mounted tricolored lamp whose purpose is to inform the driver of the following vehicle of the speed range:

- below 60 km/h : red light
- from 60 to 110 km/h : amber light
- over 110 km/h : green light

This type of permanent lamp also serves:

- at night, as a parking lamp
- as a blinker with dual intensity for day and night
- as a danger signal (symetrical blinker function right and left)

But its basic utility, from the standpoint of safety, is the assistance it brings to the following driver at high speeds. As a matter of fact, we found that the most important data when driving on superhighways was the ability of a driver to determine the speed level of the vehicle ahead from very far away so that he could take the necessary steps to regulate his own speed in advance according to the color of the light mass 100, 500 or even 1,000 m away. This is the best possible guarantee against traffic jams or chain accidents due to high speeds.

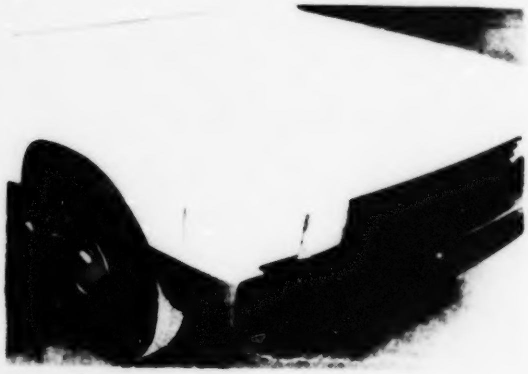

Figure 31

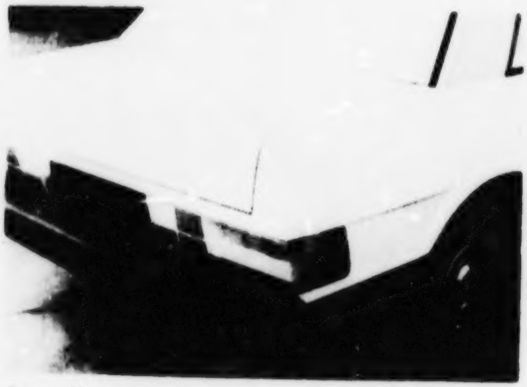

Figure 32

Finally, the ESV will be equipped with such conventional devices as:

- windshield wiper and headlamp wiper,
- rear window wiper and defroster.

CONCLUSION

We have at our disposal all basic elements for the design, materialization and testing of the Safety Vehicle. It is advisable, however, to draw up a balance-sheet for this program.

On the credit side we can put a series of partial works, mainly carried out on ESSS, with a view to a future synthetic work. This approach was the only way to cover all of the secondary safety problems within a short time and with an unquestionable efficiency.

It goes without saying that our ultimate purpose is to materialize under the best possible conditions the solutions thus obtained in our current and future production vehicles with the degree of severity justified by statistical data on road accidents.

Due to the fact that it is to be publicly exhibited, an ESV cannot resemble the prototypes developed by a manufacturer, which have to remain confidential for obvious commercial reasons.

There exist differences between the ESV and the future prototypes:

- in dimensions, even dimensions and geometric configuration of safety devices,
- in the choice of mechanical solutions and
- in producibility constraints.

There is as much a difference between the ESV and a future prototype as between this prototype and the ESSS we already realized.

Under such circumstances, is it advisable to perform extra work for a public exhibition, which is intended only for information purposes?

We wonder, today, whether we should take advantage of the studies on ESSS and pass from the study phase of the ESV to its materialization or devote our efforts towards developing a new generation of modified cars, no matter whether they are called synthetic ESSS or a partial ESV. In any case, the main point remains that the acquired knowledge and its application on future vehicles, is on a realistic basis.

SECTION TWO

PART 3

RATING ACCIDENT SEVERITIES OF OCCUPANTS

CLAUDE PROST-DAME, *General Manager*
Body Works Research Department
Peugeot Renault Association

Rulemaking concerning automobile safety (and also the statements of work for ESV) can only be based on a close observation of what really happens in road accidents. It is the only way to avoid approximate philosophies and the high cost of arbitrary requirements needed to apply them, which would lead the less-favored individuals to have no other solution for protecting themselves from danger on the road but to give up driving.

In this respect, one of the most important aims of accident investigation is finding at what speed the impact tests should be carried out. Different situations, due to the particular characteristics of traffic in different countries, may give rise to some controversy. But, in our view, the principal divergences arise from fundamental differences in the method for analyzing road observations. We are going to try and explain our ideas on the subject. To illustrate them, we are going to present the results obtained from a sample of nearly a thousand accidents in France by comparing them with other results obtained by NHTSA from American samples.

SIGNIFICANT PARAMETERS OF ACCIDENT SEVERITY FOR OCCUPANTS

To find out how to analyze an accident sample, one must first ask: How do occupants risk being killed or injured in accidents?

Firstly, the vehicle deformation has a direct influence only in as much as it alters the passenger compartment dimensions, and this constitutes braking room for the occupant (as long as he has a restraining device). We said braking room and not survival space, for the seat backs are not stiff enough to fear crushing between them and the front distorted bulkhead. In any case, non-deformation of the passenger compartment is not a sufficient condition for survival and another kinematic parameter must be found that gives the extent of impact severity.

It is not the acceleration, for in present day accidents observed on the roads, it is very rare to find belt wearers who are so tightly belted in that they can be considered as coupled to the vehicle during an appreciable part of the time the collision lasts. In almost all cases, only the speed variation of the vehicle is significant for the occupant, who is generally stabilized at his final speed when the restraining device (which is too often the bulkhead) connects him to the vehicle. In other words, it is at a speed equal to the speed variation of the vehicle that the secondary impact of occupant against bulkhead takes place if there is no restraining device. If there is one, the forces which it exerts on him are dependent on this same value. This speed must then be considered as the real criterium for the collision severity for the occupant.

CRITICISM OF PRESENT METHOD

Up to now, we have intuitively likened the speed variation to the wall collision speed that would give the same vehicle deformation. This is faulty reasoning as, in the collision with another vehicle (or even a fixed, deformable obstacle, which is often the case), there can be no independant relationship between the deformation and the speed variation of the characteristics of the obstacle collided with.

Vehicle deformation energy is only a part of the energy dissipated during the collision and is divided between the two moving bodies according to their relative stiffnesses. Furthermore, the total energy, which is equal to the total loss in kinetic energy of both moving bodies, is calculated by applying the method of retaining the quantities of movement, and thus depends not only on the closing speed but also on the respective masses of both vehicles.

Thus, to calculate the speed variation of the vehicle we must know its deformation and that of the other vehicle, as well as the masses and stress-deformation laws for each of them. These principles may be illustrated by a few simple experiments. Here are, for example, three vehicles characterized by their deformation at 50 km/h with a wall. They are an R.6 (Figure 1), an R.10 (Figure 2) and an R.12 (Figure 3). When the R.10 collides with the R.6 at a relative speed of 100 km/h, it sustains a deformation equivalent to that for a collision with a

Figure 1

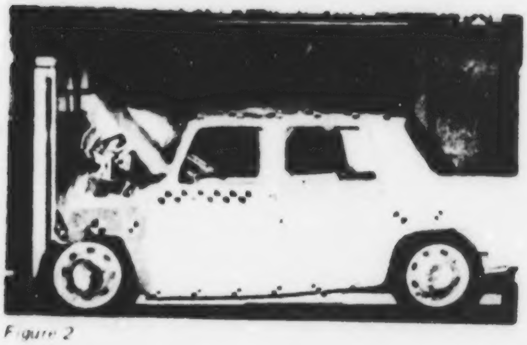

Figure 2

Figure 3

wall at 65 km/h (Figure 4), whereas the R.6 sustains a deformation corresponding to a collision at only 30 km/h with a wall. On the other hand, the same R.6 colliding with an R.12 at the same relative speed is deformed much more (Figure 5).

As the three vehicles have practically the same mass, the occupants in each one of them during both collisions would have undergone a speed variation of close to 50 km/h. Now, this method of using the equivalent deformation against a fixed barrier would have led to a faulty rating varying from 30 to 65 km/h, depending on whether one or the other vehicles is considered separately.

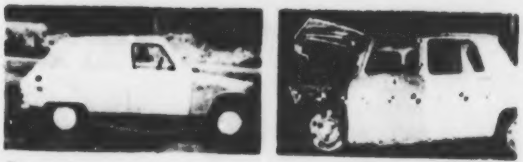

Figure 4

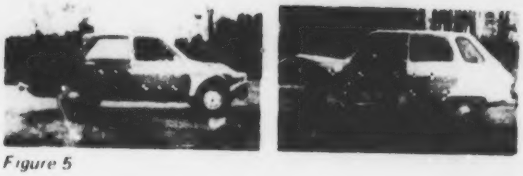

Figure 5

DESCRIPTION OF OUR METHOD

The preceding plan may be easily applied using the following calculations. If Xa Xb Ma Mb are respectively the abscissae for the centers of gravity of both vehicles and their masses, and Fab the stress at interface, we may state:

$$Ma \frac{d^2 \ Xa}{dt^2} = -Fab \quad Mb \frac{d^2 \ Xb}{dt^2} = Fab$$

By combining these equations, it becomes

$$Ma\ Mb \frac{d^2 \ (Xa-Xb)}{dt^2} = Fab\ (Ma + Mb)$$

If we call X = Xa - Xb the closing of the centers of gravity, and V = dX the closing speed, the preceding equation may be integrated in the following form:

$$\int_0^t \frac{VdV}{dt} = Fab \frac{Ma + Mb}{Ma \ \ Mb} \int_0^t dX$$

which gives $V_R^2 = \dfrac{2\ Fab\ (Ma + Mb)}{Ma\ Mb}\ Xm$

where V_R is the collision speed and Xm the closing of the centers of gravity during the collision.

Thus, the collision speed is calculated from the masses, the total deformation of both vehicles and the mean stress at interface Fab, this stress being found from knowing the stress-deformation laws for the vehicles. Knowing the collision speed, it is easy to calculate the speed variation for each vehicle by applying the method of retaining the quantities of movement: if Vf is the final speed

$$Vf = \frac{Ma\ Va + Mb\ Vb}{Ma + Mb} \text{ and } \Delta Va = Va - Vf = \frac{Mb}{Ma + Mb} V_R$$

RESULTS

We applied this method to a partial sample of accidents for which we knew the vehicle stress-

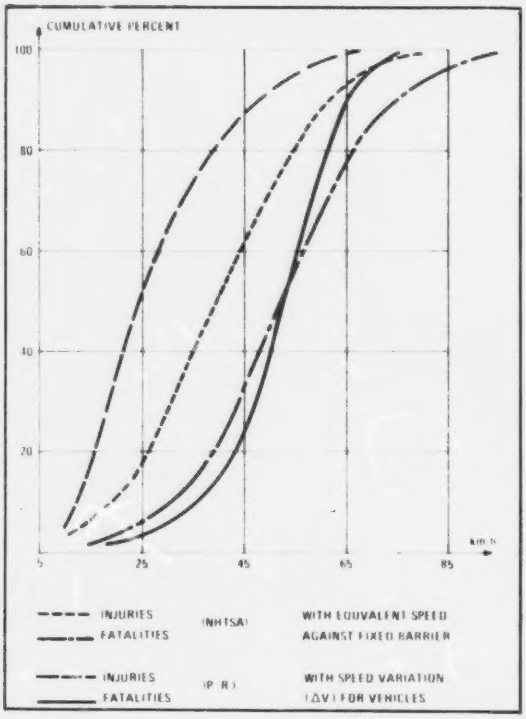

Figure 6

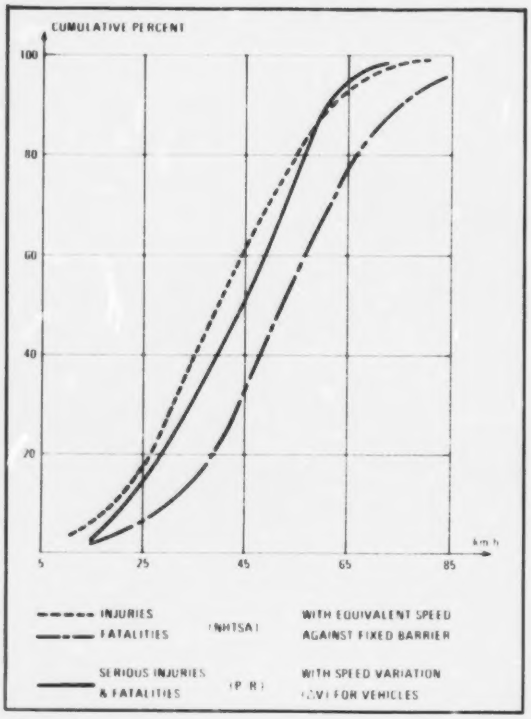

Figure 7

deformation laws. This sample was taken from that of the Association Peugeot-Renault research team. The total sample includes, at the present time, a thousand or so accidents which have been analyzed in great detail, and is representative of road accidents in France.

The number of head-on collisions in corporal accidents represents 53%, whereas for the whole of France, the national statistic gives 52%. The injury rate (killed and grievously injured in terms of the number of occupants) is also the same for the national statistics. Finally, the rating of the vehicles as a function of the NATO index is comparable to the Association sample and other local samples in France.

Our sub-sample comprises 658 injured persons, in which were involved 410 vehicles in head-on collision accidents. It does not cover only collisions between vehicles but also collisions with fixed, non-infinitely rigid obstacles (safety rails, for example).

The rating obtained is quite different from that described in the paper "Passive Protection at 50 Miles Per Hour," published by NHTSA in May 1972. Differences may exist between car populations and traffic conditions, but it seems to us that the main explanation for the divergences is the analyzing method based, in one case, on the speed variation of the vehicle, and in the other, on the equivalent collision speed with a fixed barrier.

In Figure 6, the two ratings in the form used for the NHTSA document are superimposed; cumulated percentage of killed and injured under a certain speed (or speed variation, which concerns us). We find that 93% of those injured do not undergo a speed variation greater than 50 km/h (30 mph) instead of an equivalent speed of 65 km/h in the quoted paper. Equally, 90% of those killed undergo a speed variation of less than 65 km/h instead of an equivalent speed of 78. We have added a curve (Figure 7) concerning all those killed and grievously injured, which shows that retaining devices valid for a speed variation of 60 km/h would protect 90% of them.

It may be objected that, even if the rating according to the equivalent speed against a fixed barrier is incorrect case by case, statistically it is valid, for the average of both speeds thus obtained for two vehicles colliding with each other is not very different from the mean of the speed variations calculated with the other method. For example, in the R.6-R.10 collision examined above, the ΔV's of approximately 50 km/h are very close to the mean of the equivalent speeds against a fixed barrier, which were respectively 30 and 65 km/h. But in fact, even if the mean Gauss distribution remains unchanged, the standard deviation is reduced in the exact method, and dispersion is much less, which explains why the S curve in the "histogramme" is upright and the maximum speed variation undergone by 90% of the casualties is decreased by 15 km/h.

CONCLUSIONS

We think we have shown that we can, with a certain degree of accuracy, aim at performance for the restraining devices as a function of, an exact analysis of the accidentology results. Our application of this method shows that, before envisaging higher performances, a greater protection efficiency for restraining devices that are valid for speed variations of 50 km/h may be hoped for. We think priority action should be undertaken in this direction.

This does not mean that there is nothing to be done on the structures; namely, to prevent the passenger compartment, which forms the occupant braking room necessary for the correct working of the restraining device, from collapsing. But, this must be done with some discernment while bearing in mind the problems of compatibility due to the different masses. In particular, one sole collision with a fixed barrier, which would serve for checking the efficiency of the restraining device for a given ΔV, cannot serve alone for checking the behavior of the structure. This would lead to proportioning the stiff sections to the masses and also to systematically concentrating the gradual loss of energy by deformation in the lighter vehicle.

THE FRENCH TECHNICAL PRESENTATION

SECTION TWO PART 3

PROPOSAL FOR TEST EVALUATION OF COMPATIBILITY BETWEEN VERY DIFFERENT PASSENGER CARS

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During the ESV conference in Washington, we noted that there are very few accidents today, involving frontal impact, in which an equal degree of vehicle deformation occurs.

We have seen, moreover, that the problem is further complicated by the considerable mass difference between vehicles. However, it is possible to reach a solution if it is recognized that the large vehicle can be subjected to a greater degree of deformation than the small one.

Our work on this point has continued but has been oriented towards research for parameters governing compatibility, not aggressivity. Indeed, it now seems more important to concern ourselves with solutions, rather than causes

In this work, we have distinguished two types of compatibility:

1. Structural compatibility characterizing two vehicles which, when colliding, do not deform one another to a greater degree than in the reference test or tests as regards the passenger compartment.
2. Restraining means compatibility characterizing two vehicles which, on colliding, do not mutually bring about conditions liable to exceed the performances obtained using the dummy in the reference test or tests.

The first type of compatibility is an imperative but does not guarantee protection of the occupants.

The different relations characterizing these two compatibility types can be illustrated by a graph (Figure 1) which can be used to show a compatibility velocity limit as a function of a given mass ratio for two vehicles; accordingly, the ordinate represents the compatibility velocity limit beyond which either condition 1 or condition 2 is not fulfilled.

The main hypothesis required for plotting this graph is that, if condition 1 or condition 2 is fulfilled in the reference test or tests, it is no longer fulfilled at V + ΔV, ΔV being small. This hypothesis is certainly severe, since no one can state nowadays that, if it is possible to satisfy the criterion for injuries at 50 km/h and not 52, the occupants will necessarily be killed.

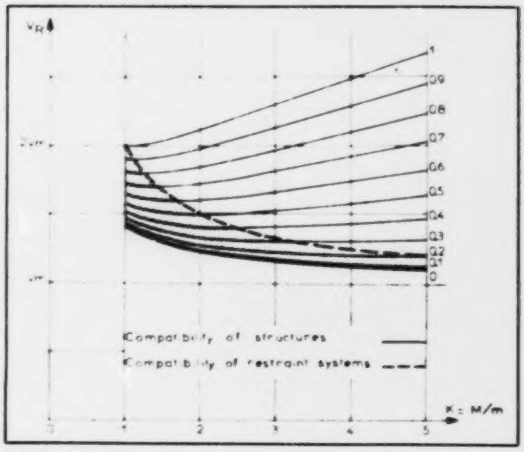

Figure 1

To take a first vehicle (i.e. A), in the fixed barrier collision test at an initial impact velocity Vm, this vehicle is compatible if:

- structural deformation does not exceed a given limit,
- the injury criteria applied to the dummies are met.

If we assume that impact takes place without elastic recovery, vehicle rebound velocity being nil, the velocity variation incurred by the occupant, which is characteristic of impact violence, is precisely the initial velocity at which the vehicle collides with the fixed barrier.

Vehicle A will be called compatible with the wall up to the velocity Vm if it meets the two abovementioned structural and restraining means requirements.

Under these conditions, if m is the mass of a vehicle A, structural compatibility can be characterized by the maximum energy W_A that structure A can absorb.

$$W_A = \frac{1}{2} m\, Vm^2$$

Note: If the impact occurs with a certain rebound velocity, i.e. β Vm the absolute value of this velocity with $\beta < 1$, structurally speaking the dynamically absorbed energy is W_A, identical with that previously defined; as regards restraining means, occupant ΔV is equal to $Vm(1 + \beta)$.

Under these conditions, guaranteeing barrier compatibility up to an initial velocity Vm means:

- as regards structure, absorbing $W_A = ½\, m\, Vm^2$
- as regards restraining means: injury criteria satisfied for a $\Delta V = Vm\ (1 + \beta)$.

Subsequently we consider $\beta = 0$.

We shall now take a second vehicle B with a mass M and suppose it also to be compatible with the wall up to the same velocity Vm.

Under these conditions, maximum energy that can be absorbed by the structure is $W_B = ½\, M\, Vm^2$.

We shall subsequently obtain $M > m$, hence $\frac{M}{m} > 1$.

STUDY OF IMPACT BETWEEN TWO VEHICLES

Structural Compatibility

Energy to be absorbed by structural deformation in the case of impact between vehicles where relative velocity is cancelled out.

We consider the previous two vehicles (A and B) travelling towards one another:

Vehicle A : mass m velocity V_A
Vehicle B : mass M velocity V_B

$$V_A + V_B = V_R$$

(V_R : relative velocity at which the two vehicles meet)

Initial kinetic energy for the two vehicles:

$$E_{CA} = ½\, m\, V_A^{\,2}$$
$$E_{CB} = ½\, M\, V_B^{\,2}$$

After impact, if we consider this as being a "soft" impact, the two vehicles will both continue moving at final residual velocity Vf. Owing to the conservation of the quantity of movement; Vf can be calculated:

$$mV_A - M\,V_B = (m + M)\, Vf$$

$$\text{i.e. } Vf = \frac{mV_A - MV_B}{m + M}$$

final kinetic energy is:

$$E_{Cf} = ½\, (m + M)\, Vf^2$$

The energy to be dissipated through structural deformation of the vehicles is equal to the difference between total initial kinetic energy and final kinetic energy, i.e.

$$W = (E_{CA} + E_{CB}) - E_{Cf}$$

$$W = ½\, (m\, V_A^{\,2} + MV_B^{\,2}) - ½\, (m + M)\, Vf^2$$

$$W = ½\, \frac{m\,M}{m + M}\, (V_A + V_B)^2$$

The energy to be absorbed on impact is:

$$W = ½\, \frac{m\,M}{m + M}\, V_R^{\,2}$$

in which m and M are the masses of the two vehicles and V_R is their closing velocity.

Minimum and maximum structural compatibility; let us take two extreme cases.

Minimum Compatibility – (only the lighter vehicle is collapsible) – The energy to be absorbed by deformation must be absorbed by the deformation of one vehicle: vehicle A.

Maximum absorbable energy for such an impact is W_A corresponding to the energy that can be absorbed by vehicle A during a test at the compatibility velocity limit for impact between vehicles is V_R, such that:

$$½ m\, Vm^2 = ½\, \frac{m\,M}{m + M}\, V_R^{\,2}$$

$$\text{hence: limit } V_R^{\,2} = \frac{m + M}{M}\, Vm^2$$

$$\text{If } \frac{M}{m} = K \qquad V_R = \sqrt{\frac{K + 1}{K}}\, Vm$$

This velocity is the maximum structural compatibility velocity for impact between vehicles in the case of minimum compatibility, i.e. in the case of impact between a deformable vehicle and an infinitely rigid vehicle.

Note: When only the bigger vehicle is crushable, V_R limit becomes $\sqrt{K + 1}\ Vm$ which is of higher value than in the preceding case.

Maximum Compatibility – i.e. when the stiffnesses and relative masses are in such a relation as each of both vehicles dissipates all the energy it is able to do.

When vehicle A collides with vehicle B at the initial closing velocity V_R, the energy absorbed by vehicles A and B is equal to the sum of the energy that they absorb in a wall impact at compatibility velocity Vm.

For vehicle A $W_A = ½\, m\, Vm^2$
For vehicle B $W_B = ½\, M\, Vm^2$

Under these conditions the compatibility velocity limit for impact between vehicles is V_R such that:

$$\tfrac{1}{2} m\, Vm^2 + \tfrac{1}{2} M\, Vm^2 = \tfrac{1}{2} \frac{m\,M}{m + M} {V_R}^2$$

$$\text{i.e. } {V_R}^2 = \frac{(m + M)^2}{m\,M} Vm^2$$

$$\text{If } \frac{M}{m} = K \qquad V_R = (1 + K) \sqrt{\frac{1}{K}}\, Vm$$

This velocity is the maximum structural compatibility velocity for impact between vehicles in the case of maximum compatibility.

General Case Two vehicles structurally compatible collide with an initial closing velocity of V_R.

The energy absorbed by vehicle A structure is

$$W_A = \tfrac{1}{2} m\, Vm^2$$

The energy absorbed by vehicle B structure is:

$$W_B < W_B = \tfrac{1}{2} M\, Vm^2$$

Let us take a velocity $V_B = a\, Vm$ (with $0 < a < 1$) such that:

$$W_B = \tfrac{1}{2} M\, {V_B}^2$$

On colliding, structure B receives an impact equivalent to a barrier impact at a velocity

$$V_B = a\, Vm \quad W_B = \tfrac{1}{2} M\, a^2\, Vm^2$$

Under these conditions, we can define a compatibility velocity V_R such that

$$\tfrac{1}{2} m Vm^2 + \tfrac{1}{2} M a^2\, Vm^2 = \tfrac{1}{2} \frac{m\,M}{m + M} {V_R}^2$$

$$(m + a^2 M)\; Vm^2 = \frac{m\,M}{m + M} {V_R}^2$$

$${V_R}^2 = \frac{(m + M)\,(m + a^2 M)\, Vm^2}{m\,M} Vm^2$$

Let us assume $K = \dfrac{M}{m}$

$$V_R = \sqrt{\frac{(1 + K)\;(1 + a^2 K)}{K}}\; Vm$$

Maximum and minimum compatibility velocities correspond respectively to $a = 1$ and $a = 0$. a is thus the *frontal compatibility factor*.

Restraining Means Compatibility

According to the compatibility definitions given above, occupant ΔV, for impact between vehicles, must be less than or equal to $\Delta V = Vm$ obtained in barrier impact, since $m < M$, $\Delta V_A > \Delta V_B$ is necessarily true.

Reminder: velocity variation for a vehicle sustaining impact The two vehicles travel towards one another at a closing velocity $V_R = V_A + V_B$

$$Vf = \frac{mV_A \quad MV_B}{m + M} \text{ hence } \Delta V_A = V_A \quad Vf = \frac{M}{m + M} V_R$$

Restraining means compatibility We must have $\Delta V_A \leqslant Vm$

$$\text{i.e. } \frac{M}{m + M} V_R \leqslant Vm \qquad V_R \leqslant \frac{m + M}{M} Vm$$

$$\text{If } K = \frac{M}{m} \qquad V_R \leqslant \frac{1 + K}{K} Vm$$

To take an example, for two vehicles with a mass ratio of 3:1, compatible for barrier impact up to 65 km/h, restraining means compatibility requires relative velocity less than 87 km/h.

Requirements are less severe for structural compatibility since a factor $a = 0.32$ is sufficient to give the two vehicles (A, B) compatibility.

Note: This graph (Figure 1) shows that, with a mass to mass ratio equal to 1, maximum possible compatibility is such that the closing velocity of the two vehicles may not be greater than twice the reference test velocity on condition that the vehicles are totally compatible. On the other hand, it can be seen also that this limit is the same in the cases of both structure and restraining means.

However, still taking the mass to mass ratio of 1, it can easily be seen that at the lower limit obtained, when one of the vehicles is considered as being infinitely rigid, the maximum compatibility velocity is, in this case, equal to $\sqrt{2} \times$ reference test speed: $\sqrt{2}\, Vm$.

On the other hand, the compatibility limit for restraining means remains unchanged, the velocity variation for the occupants of both vehicles being the same as in the reference test.

If we now vary the mass to mass ratio, we have to distinguish two cases:

What Happens Structurally:

1. If the vehicles are totally compatible, it can be seen that the compatibility speed limit will increase and thus exceed twice the reference test

velocity. This is because the total energy dissipated in a collision between vehicles of different masses is less than the sum of energy dissipated by each vehicle in a half relative speed reference test, this being since the heavier vehicle always has a non-zero final velocity in collision with a lighter vehicle, which is not the case of the reference test (the dissipated energy is proportional to the harmonic mean of the masses).

2. If one of the vehicles is totally rigid and thus incompatible, the structural compatibility speed limit will decrease in a ratio that is a function of mass to mass ratio.

What Happens With Respect To Restraining Means If the problem of keeping the passenger compartment intact is considered as settled, the compatibility speed limit for restraining means will depend only on the mass to mass ratio and thus decrease as this ratio increases.

It is interesting to note in this graph that there is a zone in which it is more profitable to make improvements structurally than with respect to restraining means. This is a zone in which many vehicles are located nowadays (shaded zone).

When the restraining means compatibility limit is reached, it is more profitable to improve its performance than to improve structural performance. Theoretically, it is only in proportion as restraining means performance level is increased that it becomes profitable and essential to improve structural performance in order to increase the compatibility speed limit.

The vital point, both as regards finding possible solutions and their cost, will be to define to what extent compatibility velocities and mass ratios must be maintained.

The cost-effectiveness ratio will obviously be closely linked to the choices taken. In order to clarify these choices, it is important to be acquainted with two essential statistical elements relating to the causes of death and serious injury in accidents.

The Compatibility Velocity To Be Aimed At The distribution curve for those killed and seriously injured as a function of the severity of the accident is needed for this purpose. The severity chiefly depends on the velocity variation undergone by the occupant and the crash duration. We found it more logical to classify the killed and seriously injured as a function of a factor depending on ΔV rather than according to a hypothetical equivalence for violent impact against a fixed barrier.

In order to select the value for ΔV giving the desired cost-effectiveness ratio, the solution cost variation as a function of the severity of the collision for the occupant must be known.

Choosing the percentage of occupants to be saved and the cost-effectiveness ratio is not the manufacturer's business.

The Maximum Mass To Mass Ratio Of Collisions To Be Aimed At The curve giving the number of collisions between vehicles as a function of mass to mass ratio is needed for this purpose.

Indeed, it is important not to take compatibility arrangements for vehicles with masses of 1 to 7, for example, as binding, if it is statistically proved that such collisions practically never occur or that their percentage is so small as to prejudice the cost-effectiveness ratio in advance.

But, just as it was necessary in the first case to evaluate cost increase as a function of improvement in performance in order to obtain the cost-effectiveness ratio, in the present case it is necessary to know how the costs of improving both the structure and restraining means of the vehicle evolve as a function of mass ratios, in other words the ΔV.

With regard to this problem of defining the compatibility limit both for structure and restraining means, it should be noted that the two parameters which will furthermore limit the range of choice that can be made as a function of statistical distribution, will be the maximum degree of deceleration tolerable for the small vehicle and the maximum collapse distance acceptable for the large vehicle.

These studies as a whole are more complex than it would seem at first since they must also take into account the transitory phase in which the car population is made up of vehicles which are not mutually compatible, but to which will be added new vehicles which must be compatible both with existing vehicles and future vehicles.

PROPOSALS FOR TESTS

After carrying out this analysis, our problem was to attempt to find the reference test that was capable simultaneously of defining:

- structural compatibility,
- restraining means compatibility,
- and, if possible, lateral impact compatibility.

Measuring the Stress Time Or Stress Collapse Relationship

This is simple, but precautions must be taken when interpreting.

The main purpose is to be sure that the vehicle will be compatible with a fixed obstacle or another vehicle for a given value of ΔV.

For this purpose, it is theoretically sufficient for each vehicle to possess a zone that can be deformed, under identical stress, and sufficient to guarantee structural and restraining means compatibility for impact with a fixed barrier.

It is certain from the foregoing discussion that, if the above condition is fulfilled, vehicles will be structurally compatible at least up to a closing velocity equal to twice the fixed barrier test velocity, the restraining means compatibility limit no longer being dependent on mass ratio.

A test which is easy to carry out could thus be based on impact with a fixed barrier at a given velocity that is a function of ΔV statistical analysis.

During this test, the compressive stresses resulting from impact with the barrier are measured, these stresses being distributed over three zones (Figure 2) (bottom middle, bottom left and right, and top) and necessarily maintained below certain limits which are the same in all cases. These limits will depend on the shortest stopping distance tolerable for the small vehicle and the longest collapse distance acceptable for the large vehicle.

During this test, the injury criteria are measured.

In order to be certain that this gives the best results, it is insufficient to require stresses to be below a certain limit, it would seem preferable to aim at locating them in a given range.

Structural diversity greatly detracts from the simplicity of this test. Indeed, thin sheet metal components are deformed absorbing a deformation energy which is dependent on the force required to attain this degree of deformation and the distance over which this stress is maintained.

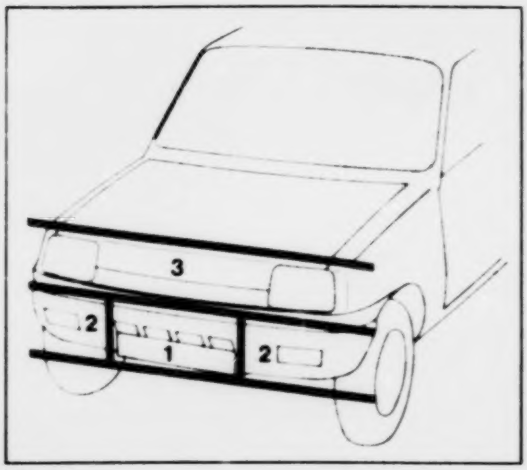

Figure 2

When the structure of the vehicle opposite cannot withstand this stress, it is the one to be completely deformed. On the contrary, the incompressible intermediary masses located within this structure exert a stress on the obstacle depending on the value of the mass, but also on the rigidity of the obstacle, which gives very high values in the case of a rigid wall.

When the intermediary mass is travelling at the same velocity as the obstacle, its action ceases and it no longer applies any stress upon it.

In the case of a collision between vehicles, this means that the action of the intermediary masses is, as regards the peak stress, below what is measurable against a fixed barrier.

The test method described above could thus be unnecessarily severe for a vehicle, far exceeding the limit imposed, without its necessarily being incompatible, if only a stress peak caused by the arrest of an intermediary mass is involved.

Those objections could be discarded by measuring both the power unit acceleration and the three partial stresses described above, a computing program then being used to deduct automatically for the part of the power unit.

In order to show not only the advantages but also the restrictive and confining characteristics of this method, we have made a calculation using a mathematical model with two masses and three degrees of rigidity (Figure 3).

Good repeatability was obtained between tests and calculations by simulating the vehicle with a power unit mass and a passenger compartment mass as well as rigidity between the compartment and the obstacle, between the power unit and the obstacle and between the power unit and the compartment (Figure 4).

Measurement of stress in several zones is not redundant with the deceleration measurement, for only the first type can enable the degree of rigidity between the power unit and the compartment to be

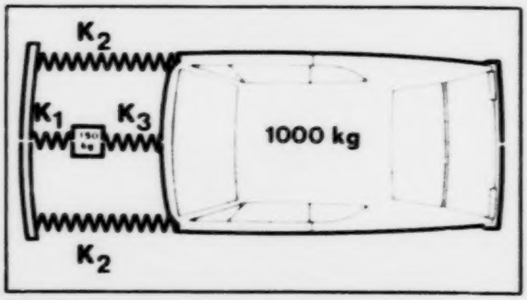

Figure 3

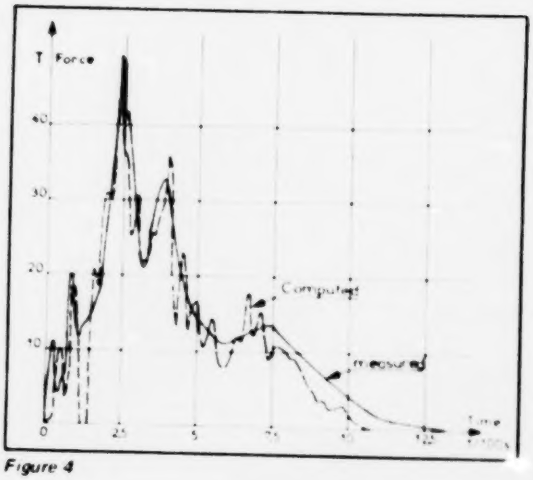

Figure 4

determined differentially when the power unit mass is stopped, the transversal connections being neglected.

In order to illustrate the numerous calculations made, we have collected the following typical examples:

1. vehicle to be studied, constituted of two masses and three types of rigidity (Vehicle S, Figure 5).
2. a vehicle without power unit and only one equivalent type of rigidity (Vehicle A, Figure 6).
3. a vehicle with two masses identical with (1) but without connection between the power unit and the passenger compartment (Vehicle B, Figure 7).
4. a vehicle with two masses identical with (1) but without rigidity ahead of the power unit (Vehicle C, Figure 8).
5. a vehicle with three types of rigidity and a compartment mass identical with (1) but doubled power unit mass (Vehicle D, Figure 9).

Figure 10 summarizes the results, which indicate:

- That there is, in fact, no direct relationship between maximum stress on impact with a fixed barrier and aggressivity.
- That if the vehicles have adjacent stress levels, structural compatibility can be guaranteed however this stress is set up.
- That the fixed barrier cannot characterize the behavior and variation in collapse for different vehicles colliding with one another. In addition, since it is proposed to measure the injury criteria in the same test, these criteria must be measured at a precise velocity that cannot be modified in order to vary the degree of collapse.
- That these calculations bring out other possible methods such as collision with an energy absorber, whether or not its mass is infinite. In this case, the problem is then to define a type of absorber which is satisfactory both for the large and the small vehicles. It is then possible to design two tests at different velocities or involving impact between different masses in order to cover all collisions with fixed or mobile obstacles.
- That it seems difficult that an aggressivity index will cover the complex range of all cases of frontal impact and at the same time incorporate the notion of lateral impact compatibility.

The first test in measuring forces and injury criteria up to a given maximum level must be supplemented by another test at higher speed in which forces must be higher than a given value;

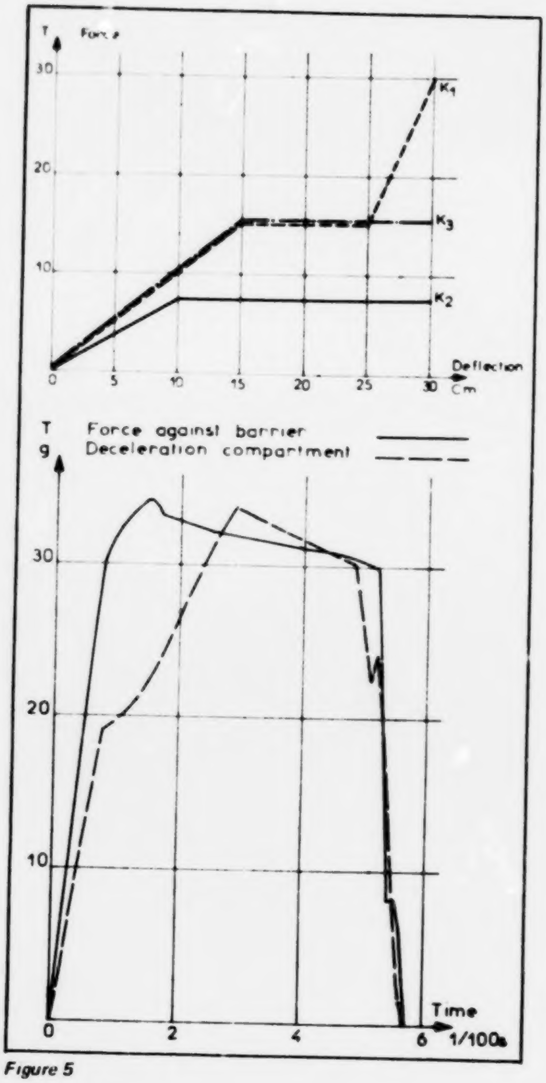

Figure 5

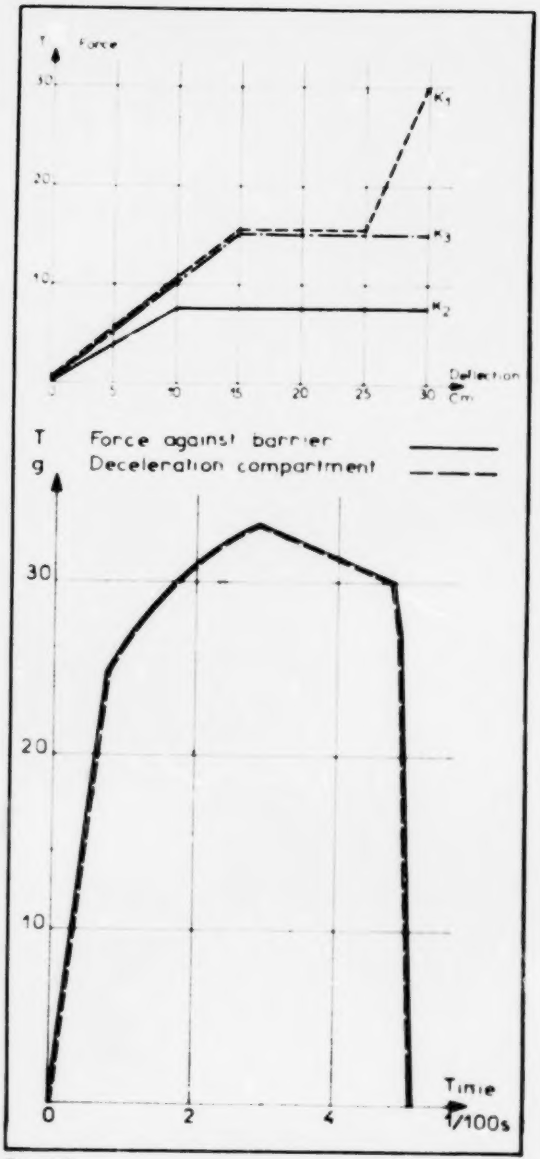

Figure 6

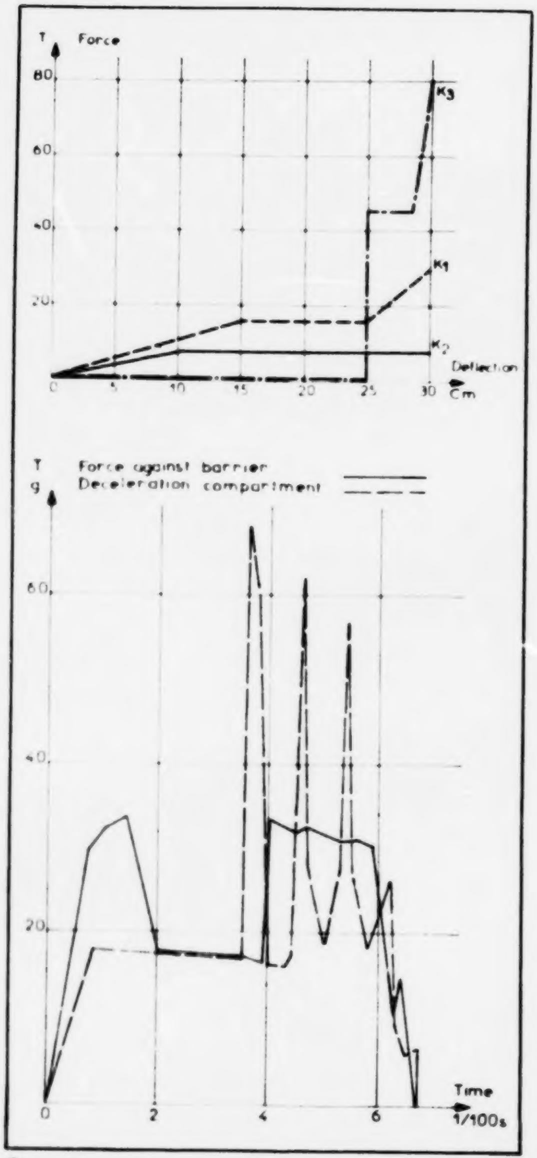

Figure 7

injury criteria are not to be measured during this second test.

The maximum force level chosen for the first test at Vm_1 is used to check that the vehicle crush length is sufficient to allow:

1. Kinetic energy to be absorbed without exceeding a standardized force level.
2. Fulfilling injury criteria.

The force level chosen for the second test at $Vm_2 > Vm_1$ is used to check that the vehicle is able to withstand a force level higher than in the Vm_1 test, but only beyond crushing distance corresponding to the first test.

This step of minimum force level is also standardized regardless of vehicle weight of type.

In a head-on collision involving two cars both fulfilling the two conditions above, one can be sure to dissipate a maximum quantity of energy before reaching the limit of one of the vehicles, and to use the compatibility limit up to a maximum even where vehicles are of vastly different weights, and sizes.

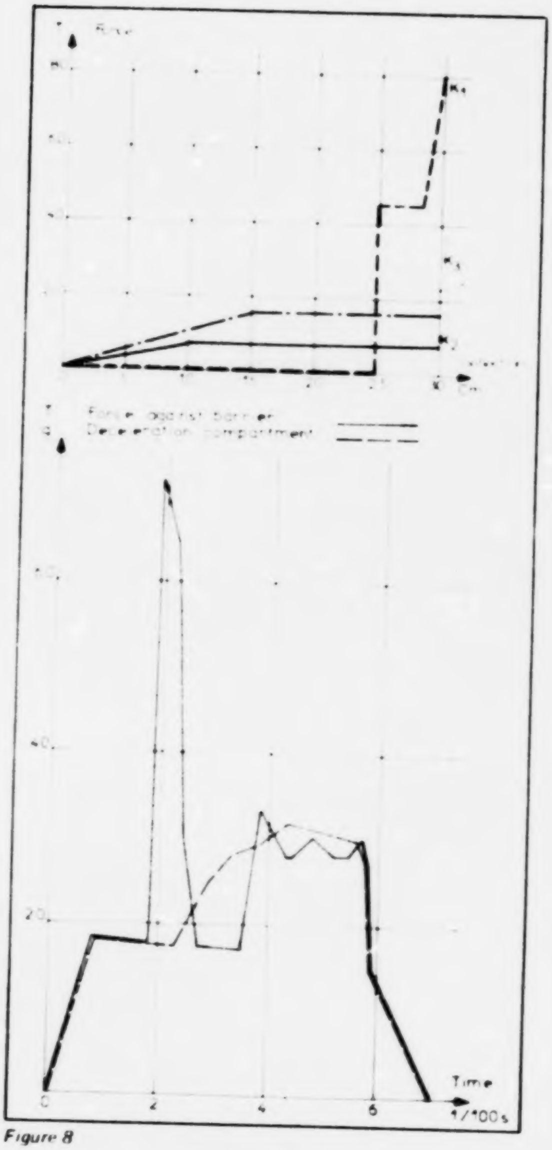

Figure 8

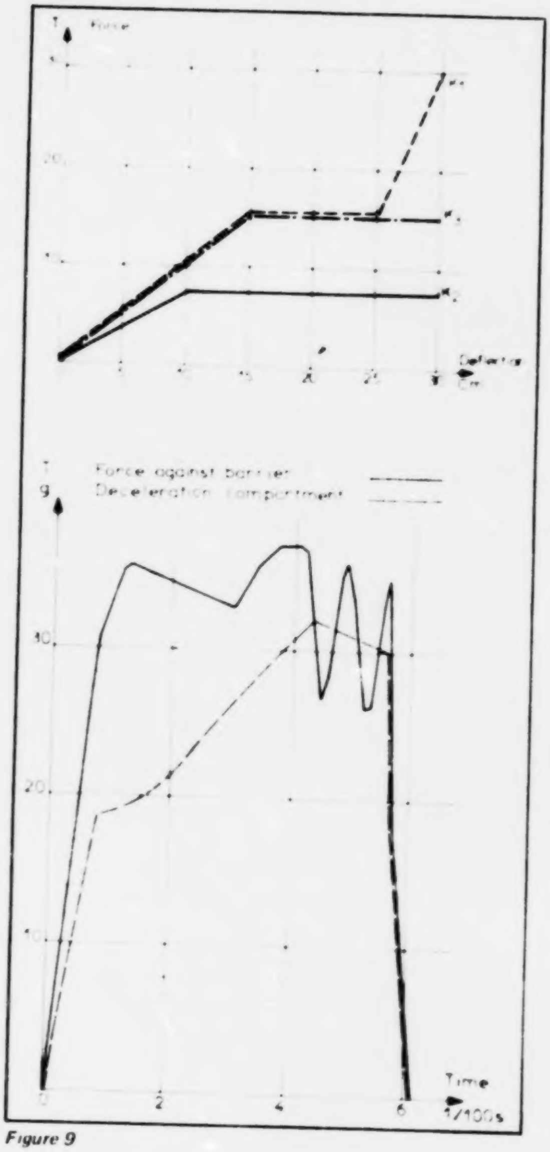

Figure 9

This method requires that the problem of intermediate masses is solved beforehand and cannot be applied without adaptation to cars fitted with hydraulic bumpers.

Other Possible Methods

In view of the restrictions imposed on the above method, it is possible to invisage integrating intermediary mass effects by evaluating the degree of collapse that they communicate to an energy absorber applied against the wall.

It is the degree of absorber penetration, limited to within certain maximum and minimum values, which would guarantee compatibility for the vehicle tested with other unknown vehicles which nonetheless satisfy the same requirements.

The drawback to this measuring method is that it does not take into account lateral collision problems.

The above method may be further complicated by replacing the single absorber by four absorbers, three for the bottom and one for the top, calibrated at different values and giving a maximum and minimum degree of collapse for each zone.

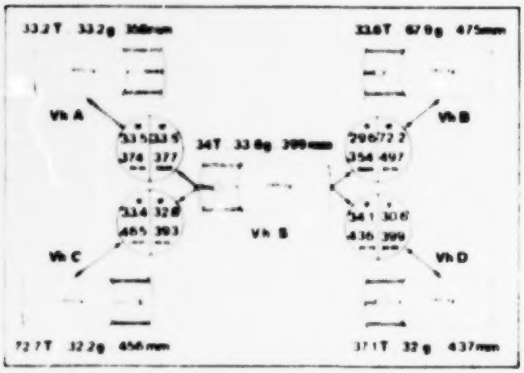

Figure 10

This theoretically solves the problem of evaluating lateral impact compatibility, but rather complicates the test procedure.

Impact Against A Mobile Barrier

We saw above that it was quite possible to consider evaluating compatibility using a test for impact with a mobile barrier. But it is obviously of no interest to carry out this test using a rigid mobile barrier.

Mobile Barrier With A Single Slowing Device The above argument for impact with a fixed barrier also applies here but two cases have to be distinguished:

- *Mobile barrier mass greater than that of the vehicle tested.* A single test is sufficient in this case since the ΔV sustained by the occupant will be greater than vehicle-barrier relative half velocity, a velocity which would correspond to that for impact with a fixed barrier.
- *Mobile barrier mass smaller than that of the vehicle tested.* A single test is no longer sufficient to guarantee vehicle compatibility for a fixed obstacle, or a different impact velocity has to be considered, which again brings up the problems of calibrating the absorber.

In all cases, the single slowing device does not meet lateral impact requirements.

Mobile Barrier With Multiple Slowing Devices While lateral impact problems are resolved in this case, the comments made in the previous case hold good. In all cases of mobile barrier testing, it is also possible to envisage multiple testing at different velocities or using mobile barriers of different masses, but this again complicates the methods.

CONCLUSION

We have seen that equal chances of survival for the occupants of broadly differing vehicles in the event of mutual collision depend on solving complex problems and that the chances will never be completely equal.

However, extensive analysis of statistics and theoretical and experimental studies lead us to believe that it is not impossible to arrive at broad lines of research.

This difficult problem which is further complicated by the desire to cover lateral collisions at the same time, results in the use of many different test methods for characterizing vehicle compatibility with fixed and mobile obstacles and varying masses and constitution.

None of these methods is perfect and much remains to be done before making a selection.

However, we should not wait too long before making an effort in this sphere either by general agreement between manufacturers or by introducing regulations.

Since these ideas can only be applied at the design stage, and in view of design deadlines, these ideas can only be applied in the remote future owing to the time taken to evolve them.

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THE FRENCH TECHNICAL PRESENTATION

SECTION TWO

PART 3

STATEMENT OF CITROEN PROGRAM

MAURICE CLAVEL, *General Delegate*
Anonymous Society of Citroen Automobiles

The Citroen Automobile Company is going to present to you, briefly, the second phase of the studies and experiments carried out in accordance with the Experimental Safety Subsystems Program (ESSS), set up two years ago by the French Government, and established within the framework of the international decision taken by the NATO Committee for the Challenge of Modern Society (NATO CCMS). The work concerns a small car designed primarily for urban use, with a maximum weight of 1,500 pounds and a length equal to, or less than 3.60 meters (about 12 feet).

The program, sponsored by the French Government, originates, as you know, from a fundamental study of the subsystems which make up the vehicle and, particularly, its structure and body, in order to improve its international performance in the case of impact and collision.

The program may not necessitate any spectacular exhibition — which is not, incidentally, the least of its merits — but it nonetheless shows the advantage of adopting a rigorous methodology, an essential factor in the insufficiently known field of the mechanics of collision. We have chosen for our study a small vehicle at the bottom of the European range which poses problems of the absorption of energy, particularly difficult to solve just because of its small size. The systematic study of subsystems is probably that which best allows us to improve the safety of the occupant in such a car without distorting its function — i.e., without removing it from its category by increasing the weight and the price which would be unacceptable to the socio-professional strata of the population for whose use it is designed.

The second phase of our study took place after deepening the contacts between our Research and Development Department and Cornell Laboratory. Nonetheless, in the same way that a golfer who is in the rough wonders what the distance is to the green that he cannot see, our researchers and our technicians still find themselves confronted with unknown problems which cause them doubts and fears, since the goal of their work is life or death.

- How far does the anthropomorphic dummy reproduce the behavior of a human being in a collision?
- How far can one submit occupants of a vehicle to stress and dynamic deceleration in a vehicle equipped with restraint devices, without their suffering grave or irreversible damage?
- What is the frequency rate of types of accidents, the index of impact speeds, the angles of incidence, and of the objects likely to hit, or be hit, by a vehicle which would permit the orientation of our selection of questions to be studied, and the improvements to be made on vehicles, in order to obtain for them the best cost/efficiency ratio, about which, as you will understand, the Common Market Commission of Constructors is worried, and with reason?

For these questions there are no precise answers and it is in spite of the problematical character of the above parameters that our studies are being followed.

Following the report which Mr. Bohers is going to give you, and at the end of the work of the second phase of studies and experiments, you will notice that it is apparently unrealistic to hope that a short vehicle will be able to undergo, with success, the tests of frontal collision against a fixed barrier that the American ESV specifications impose on a vehicle with a total weight of 4,000 pounds.

Therefore, in spite of the improvements we shall make on such a small vehicle, it will necessarily have to be the object of special regulations to assure that its occupants have an acceptable survival space with adequate deceleration of the head and the thorax.

Without such precautions, the small vehicle at the bottom of the European range will be condemned to disappear, while it is, in Europe, the major element of democratic motorization and gives the real solution to the worsening problems of urban circulation and particularly to air pollution in the big cities, as William Ruckelshaus has just reminded us in the 52nd Conference of Highway Research Board on January 24.

It is imperative that we take into account this crucial alternative.

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THE FRENCH TECHNICAL PRESENTATION

SECTION TWO

PART 3

PROGRESS REPORT ON THE SECOND PHASE OF RESEARCH AND EXPERIMENTATION ACTIVITIES

SERGE BOHERS, *Coordinator for Safety Regulations*
Anonymous Society of Citroen Automobiles

INTRODUCTION

Our company's progress report chiefly relates the sequence of work that we find essential, after our preliminary stage, for reaching the goal set by the "action thematique programmée" (E.S.S.S. French program). The second stage in our report relates to structure design. The first part concerns our research toward possible improvements to the structure of the sub-subcompact Citroen vehicle, with a view to improving its performances in frontal impact and in side impact. In the second part, we begin to analyze the behavior of passengers restrained by an active restraint system.

During the preliminary stage of our work, some of the elements which were adapted in our second stage, comprised general research into vehicle behavior in the following cases: perpendicular frontal impact against a fixed barrier, perpendicular side impact between two vehicles.

- Research of a laboratory reproduction of deflections: using a vertical drop test rig, using a pendulum drop weight test rig.
- Technological study with a view to: avoiding the buckling of structural elements, finding a "logical sequence of component deflection" for the whole of a structure.
- General checking on the vehicle enabling, particularly in the case of perpendicular frontal impact: an after-crash inspection of residual volume improvement; a measure of structure decelerations as a function of time, given that the manufacturer is aware that residual volume must represent a "survival space" for the occupants restrained by active restraint systems.
- We know that vehicle behavior in a collision depends on its structural design, on its rigidity and on its load.

COMMENTS

This preliminary research stage showed that, while impact against a fixed barrier can be validly reproduced, it is doubtful whether it is representative. French statistics show that more than 60% of persons involved in an accident are the victims of collisions between vehicles.

Our standard checks were carried out on a sub-subcompact vehicle with the following characteristics:

- overall length 3.870 m
- distance from H point to front bumper 2.040 m
- curbweight 600 kg
- maximum load 330 kg

The European sub-subcompact vehicle we have in mind must be designed with the following dimensions and price requirements:

- overall length 3.600 m with a structural design such that:
- distance from H point to front bumper is 1.800 m

Having defined our objective of improved safety conditions for occupants restrained by active restraint systems, we seek answers to two major questions:

- Up to what speed?
- For how many occupants? and also
- At what price?

WORK PROGRAMS

Perpendicular frontal impact against a fixed barrier

- structure, influence of:
- impact velocity
- vehicle load

Criteria:

- geometrical
- dynamic

1. Resumption of testing with a standard type Dyane vehicle.
2. Vehicle meeting our "E.S.S.S. Program."

Perpendicular side impact

- structural alterations

ANTICIPATED PERFORMANCES

Perpendicular frontal impact, with an impact velocity exceeding 50 km/h against a fixed barrier with two or three restrained anthropomorphic dummies, improve:

- crushing strength
- distribution of buckling loads between structure

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lower and upper levels

- vehicle deceleration characteristics
  - avoiding over-long peaks
  - within a maximum tolerable by the occupants

Perpendicular side impact, for a collision velocity of approximately 32 km/h with two restrained anthropomorphic dummies, improve:

- intrusions into the passenger compartment
- protection of impact zones corresponding to a vulnerable part of the body, particularly the head

Here again, statistics show that the risk of death or serious injury is greatest in side-on collisions.

CONCLUSION

The second part of our progress report determines technically-feasible performances, enabling a sub-subcompact vehicle to be better designed from the safety point of view without prejudice to the two basic specifications weight and price.

In our present state of knowledge, and despite the restrictions of active restraint systems, we should realize that this is the only way of rapidly and radically minimizing serious consequences to vehicle occupants, in the event of a collision with another vehicle or any other obstacle.

STANDARD TYPE DYANE

Structural Test Program

Figure 1 summarizes our testing.

- Curbweight: the vehicle weighs 600 kg
- Laden weight testing: three Sierra dummies totalling 225 kg were installed and positioned in accordance with ISO DIS 2958 recommendations.

DYANE			
	Vo in km/h		
	50	65	75
CURB W.	X	X	X
LOADED	X	X	

Figure 1

Measurements Recorded

For each test at curbweight:

- Vehicle velocity at 5 m from the barrier, from which we infer the impact velocity.
- After impact, the vehicle permanent length reduction.
- During impact, deceleration of the vehicle at a point located close to its center of gravity, and in a zone of maximum rigidity.

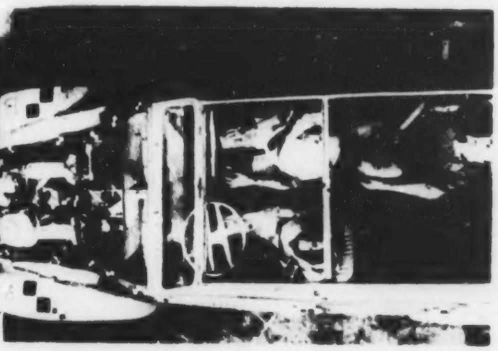


Figure 2

For each test at laden weight, three dummies were installed (driver, front passenger, one rear passenger) (Figure 2); they were restrained by a standard, active-restraint system (3-point safety belt) for the 50 km/h test, the system being duplicated for the 65 km/h test.

- On the vehicle: some parameters are above.
- On the dummies: according to the requirements of Standard 208.

  Head triaxial acceleration.

  Thorax acceleration.

  Load on femur.

Figure 3 summarizes the results:

DYANE			
	Vo km/h	ΔHT cm	γ AVERAGE g
CURB WEIGHT	50	59	16.7
	66	86	20.2
	74	99	21.8
LOADED	51	64	16
	68	98	17.5

Figure 3

- Average acceleration (average γ), expressed as a multiple of g, is calculated on the basis of permanent length reduction (ΔHT) and as a function of velocity (Vo) at impact (Figures 4 and 5).

Figure 6 emphasizes the influence of impact velocity on vehicle crushing. Curves I and II summarize the results of these first test sequences.

Figure 7 shows the vehicle deceleration rate as a function of crushing. It is obtained from the vehicle deceleration, measured during testing, as a function of time. The curve characterizes the vehicle and

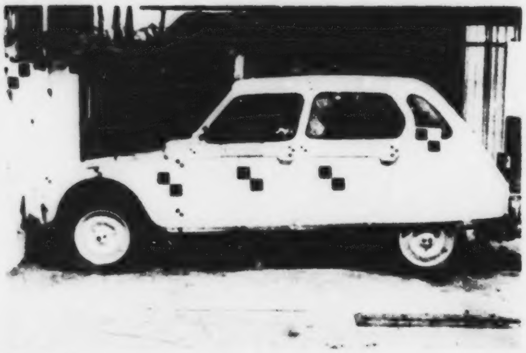


Figure 4

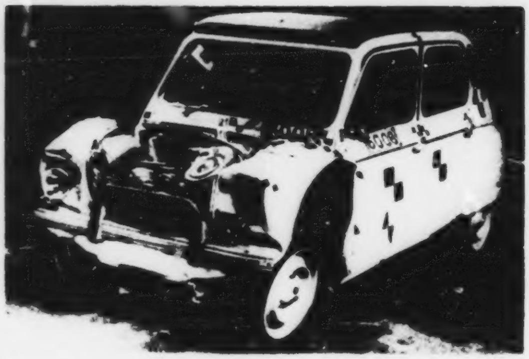


Figure 5

should enable us to improve the structure by improving its weak points.

Analysis Of Results

Effect of velocity (Curve 1, Figure 6)

- Crushing increases linearly as a function of velocity at impact.

Effect of load (Curve II, Figure 6)

- Sections a, a1 and b, b1 show how the load increases the rate of crushing.
- This effect increases with the initial velocity at impact. Indeed, since the kinetic energy of the dummies is partly absorbed by the structure through the restraint systems, this energy increases by the square of initial velocity.
- If we take vehicle crushing and residual volume as criteria, points a2 and b2 of Curve I correspond to points a and b of Curve II. We find that, with the same value of ΔHT, the divergence between velocity at impact, laden and at curbweight, is greater at 66 km/h than at 50 km/h.

  Load effect on average deceleration as a function of velocity.
- In Figure 3 the presence of dummies decreases the average deceleration of the vehicle.

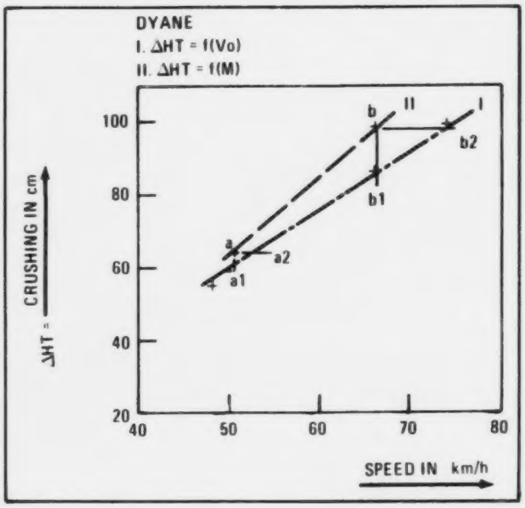


Figure 6

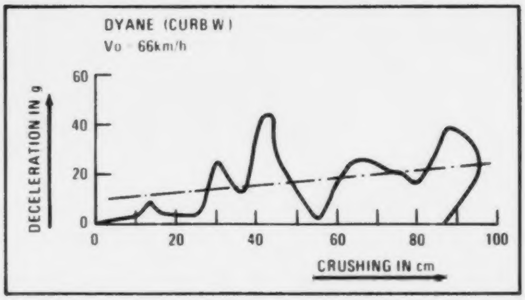


Figure 7

- Their effect is greater at 66 km/h than at 50 km/h.

DYANE ATP – DEFINITION

The Dyane ATP is a vehicle derived from a Dyane sedan incorporating structural alterations to improve performance in the event of a frontal impact against a fixed barrier. We are stating the following hypotheses:

- The Dyane ATP, carrying three dummies restrained by an active restraint system, is to sustain, at a 65 km/h perpendicular frontal fixed barrier impact, a rate of crushing equal to that of the standard Dyane, similarly laden, at a 50 km/h impact.
- The effect of the dummies on the crushing value is similar in both types of vehicles.

Anticipated performance:

- vehicle at curbweight
- Vo = 65 km/h
- ΔHT = 50 cm

Characteristic curve

Figure 8 shows the vehicle's average deceleration rate as a function of crushing. Let us note that for a 65 cm crushing due to an impact, the vehicle deceleration rate will increase from 20 g to 40 g.

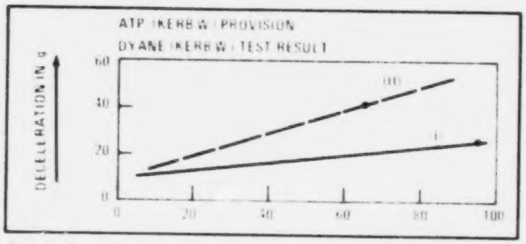


Figure 8

CRITICISMS OF THE STANDARD STRUCTURE

The characteristic curve shape in Figure 7 shows weaknesses in the standard structure:

- Up to a 25 cm crushing, the deceleration curve recorded during testing is well below the average value.
- The engine unit comes to "a stop" after a 42 cm crushing.
- With a 55 cm crushing, the platform buckles to the rear of the front axle cross member.

Structural alterations to be provided for:

- At front of vehicle
    - bumper reinforcement
    - improvement to the working of platform side members by preventing buckling and reinforcing the cross section
- In passenger compartment
    - preventing platform buckling
- Preventing engine unit from suddenly stiffening the structure, either:
    - by ejection of the unit under the crossmember,
    - by means of a sufficiently rigid ramp, or
    - by increasing absorption of kinetic energy before the unit comes into contact with the axle cross member

IMPROVEMENT RESEARCH WORK

Our preliminary stage has enabled us to test different types of shock absorbers;

- mechanical shock absorbers located forward of the side members
- injection of rigid polyurethane foam into the platform

For the Dyane ATP vehicle research, we have selected a solution to controlling platform deflection with mechanical shock absorbers (Figure 9).

When subjected to a compressive load, a slender beam tends to buckle (Figure 10). When applied to the Dyane ATP vehicle, the tube, integral with one end and guided at the other end into a hollow body, does not play any part in the compressive load. However, by being guided, it is made integral with the body for flexion and buckling.

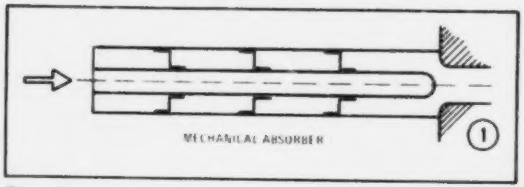


Figure 9

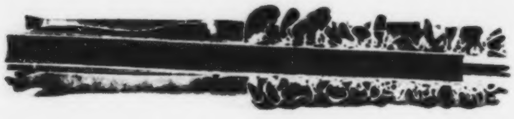


Figure 10

This system was developed during a sequence of press tests (Figures 11 and 12). The test shown was carried out using a part having the same dimensions as a platform side member.

Figure 13 shows that the load remained substantially constant, 3,6 x 10^3 kg.f over a crushing distance of more than 30 cm. During frontal impact testing, we observed that the rear of the vehicle was considerably lifted rearing which is all the more marked because the vehicle was laden.

It is difficult to give objective figures to describe the effect of this phenomenon upon occupants; however, subjectively, everything induces us to decrease this rearing. In our ATP research, we provided for structural elements which tend to raise the application point of the resultant loads.

For reinforcement of front end structure (Figures 14 and 15), the side members are extended 20 cm forward and will work in cooperation with the bumper. These side members will be constituted as mechanical shock absorbers, as described above.

The side reinforcements (Figure 16) are intended to withstand the loads transmitted to the waistline level.

Rearing is reduced by displacing the application point of applied loads (Figure 15).

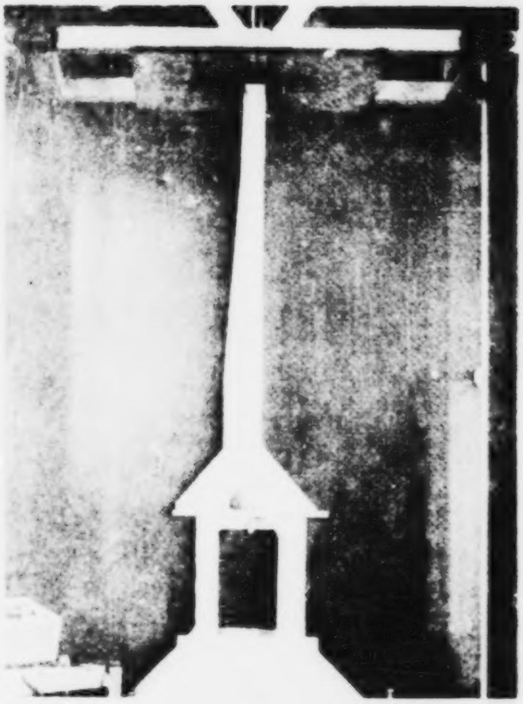


Figure 11

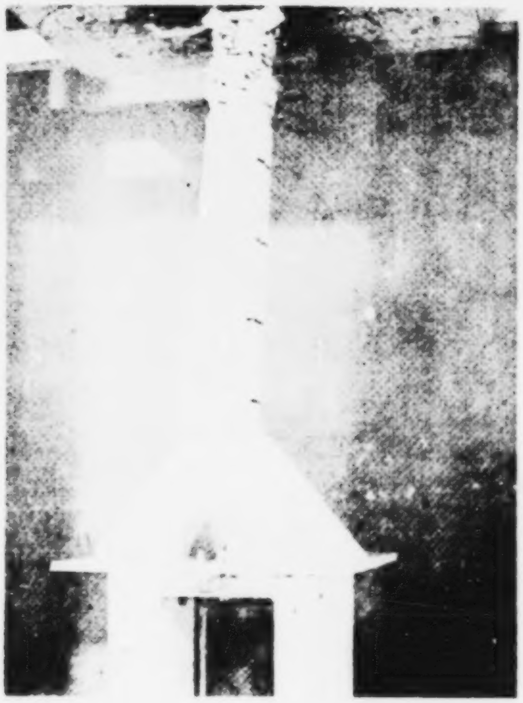


Figure 12

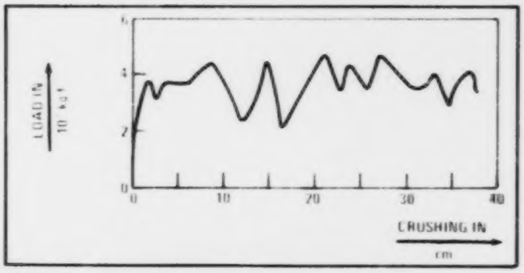


Figure 13

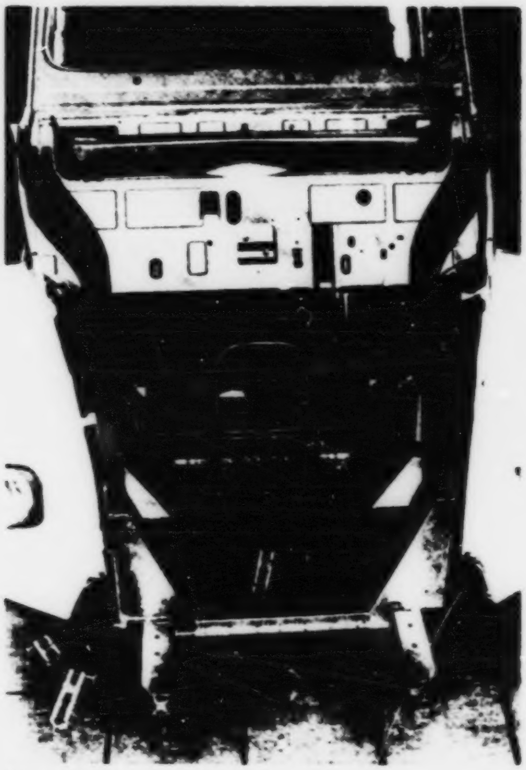


Figure 14

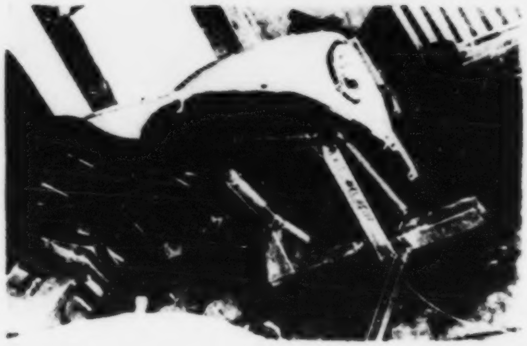


Figure 15

Figure 16

- Adjunction (to the upper part) of hollow bodies working in parallel with the platform side members, by distributing a greater percentage of the loads due to buckling towards the waistline of the vehicle.
- A tubular cross member joins these two hollow bodies to prevent them from buckling outwards under the compressive load they have to undergo (Figure 17).

We shall be required to provide body bottom side members with a high degree of inertia for side impact. We attempted to make them contribute to resisting frontal impact loads (Figure 18).

Side door reinforcement at waistline level (Figure 19) aids in transmitting loads at the waistline level and also reduces side door deflection in the event of side collisions.

Front axle cross-member extension by means of sheet metal components will load the side members through the front wheels and tires during the engine unit rearward displacement; this advantage holds good for an angle of wheel lock less than approximately 15° (Figure 20).

The effect of the structural improvements on weight is shown in Figure 21.

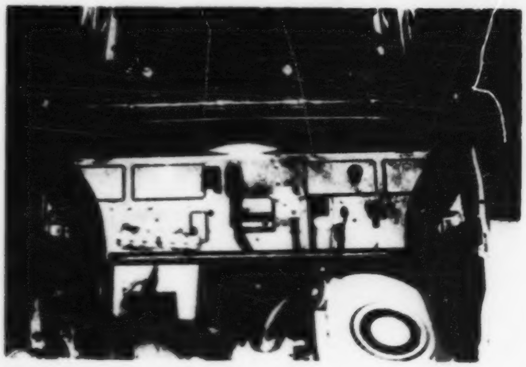


Figure 17

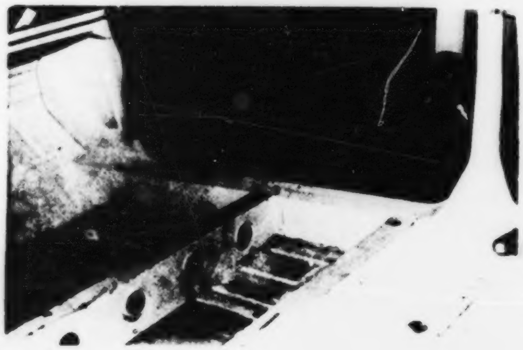


Figure 18

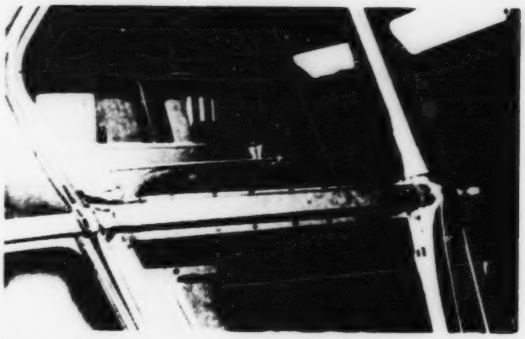


Figure 19

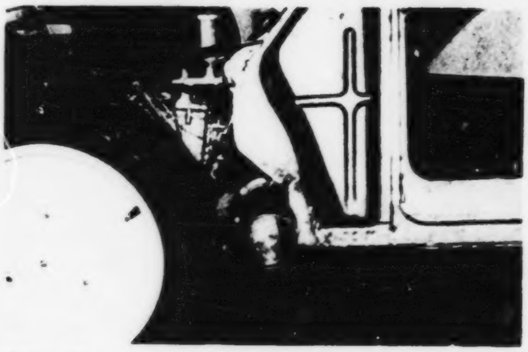


Figure 20

STRUCTURAL IMPROVEMENTS EFFECT ON WEIGHT

A T P COMPARISON		
	WEIGHT	
COMPONENT PART	**kg**	**%**
PLATFORM	+ 8	+ 11
IRON FITTED BODY	+22	+ 34
DOORS	+ 6	+ 35
FRONT BUMPER	+ 5	+170
COMPLETE BODY	+41	+ 25
VEHICLE		+ 7

Figure 21

CURBWEIGHT TESTING

Tests at curbweight were carried out on the vertical drop rig, applying the successive drop method. The effect of initial velocity on structural buckling is more easily emphasized (Figure 22).

These were carried out on the vertical rig and for each test we measured:

- impact velocity
- vehicle deceleration
- length reduction

Using a single prototype, we characterized in four successive tests:

- the modified structure as a whole
- the velocity effect

Vehicle characteristic curve (Figure 23):
$\gamma = f(\Delta HT)$, which was obtained by applying the method described in the appendix.

Permanent crushing values, as a function of drop velocities allowed us to obtain Curve III (Figure 24).

A summary of these tests is shown in Figure 25.

A.T.P.			
	Vo in kilometers per hour		
	50	65	75
CURB W	X	X	X
LOADED	X	X	

Figure 22

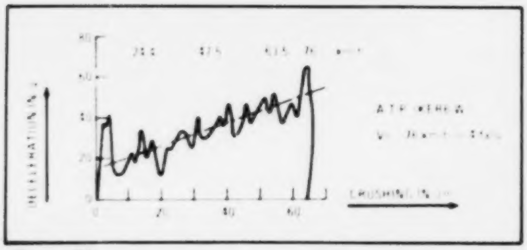


Figure 23

Analysis Of Results

A comparison of results with performances anticipated in our ATP definition is shown in Figure 26. The characteristic curve is shown in Figure 23.

The average value is slightly above the value anticipated in Curve III of Figure 8 in our ATP definition. We observe a peak for approximately 5 cm crushing; nevertheless, buckling remains localized at the very front and we observe no anomaly in the logical sequence of structural buckling.

Buckling uniformly progresses rearwards as impact velocity increases. With a 65 cm crushing, i.e. 76 km/h, acceleration reaches a peak; there are no more elements to be crushed and buckling appears in the body bottom. We have reached a threshold beyond which any buckling will affect the residual volume of the passenger compartment.

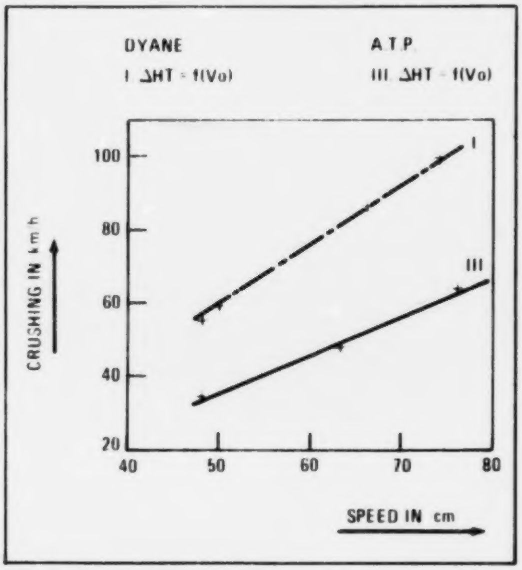


Figure 24

A.T.P.		
CURB WEIGHT TESTS		
Vo IN km/h	ΔHT IN cm	γAVERAGE IN g
48	34	26.8
63.6	48	32.4
76	64	35.5

Figure 25

CURB WEIGHT VEHICLE		
	OBJECTIVE	REALIZATION
Vo	65 km/h	63.6 km/h
ΔHT	50 cm	48 cm

Figure 26

In Figure 27 we noted a considerable increase in the deceleration rate, indicating problems in designing restraint means which will satisfy the severity indices, particularly head-thorax ones.

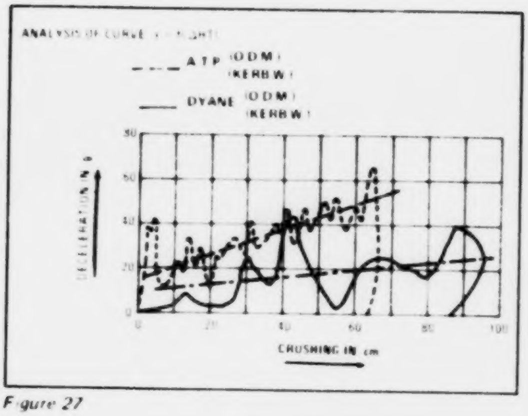


Figure 27

LADEN WEIGHT TESTING

During tests conducted against a fixed barrier three Sierra-type dummies (Figure 28) were installed (driver, front passenger, one rear passenger) and were restrained at 55 km/h, by internal shock absorption belts. For tests conducted at 65 km/h, we again selected a standard type Dyane solution, enabling the dummies to be restrained without risking strap rupture. The system adjustment for an impact velocity greater than 55 km/h is presently being carried out by means of an impact-sled.

We measured the same parameters for the vehicle and the dummies, as in the standard type Dyane test. Permanent crushing values as a function of impact velocities (Figures 29 and 30) allowed us to obtain Curve IV (Figure 31). Figure 32 shows the summary of these tests.

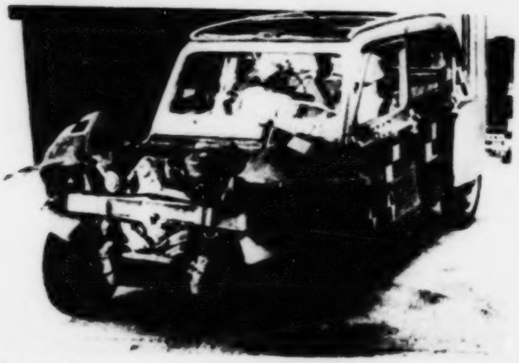


Figure 28

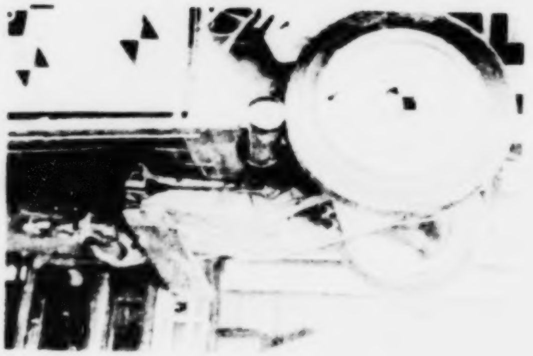


Figure 29

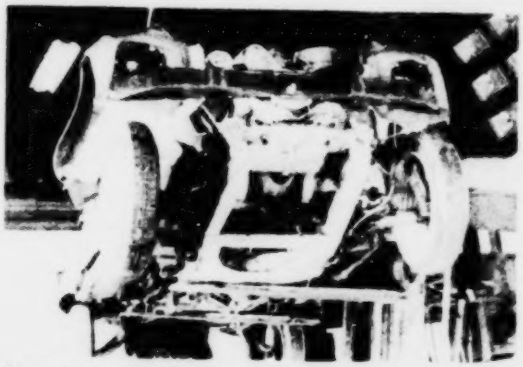


Figure 30

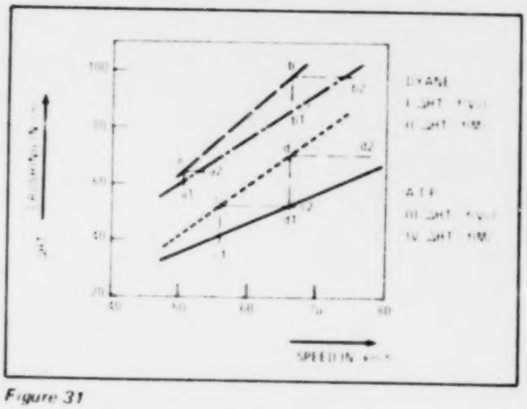


Figure 31

A.T.P.

LOADED TESTS

Vo IN km/h	ΔHT IN cm	γAVERAGE IN g
55	48	25
66.8	64	26.8

Figure 32

Analysis of Results

Effect of velocity (Curve III, Figure 31) crushing increases linearly as a function of velocity. Same findings as for the standard type Dyane (Curve I).

Effect of load on crushing (Curve IV, Figure 31) sections cc_1 and dd_1 illustrate the increased crushing of the vehicle caused by the presence of the dummies. This effect increases with initial impact velocity. An improved connection between the structure and the restraining means tends to increase the deviation in the case of the Dyane ATP vehicle. The deviation between Curves I and II, III and IV is characteristic of the connection.

For the load effect on deceleration (Figure 33) we have superimposed the characteristic curves for the vehicle at curbweight and laden. The effect of the dummies is perceptible after a 30 cm crushing. At curbweight and laden, the structure crushes under the same load; the dummies exert a stress load on the structure (through the connection); the vehicle deceleration decreases, resulting in an increase in time, and distance travelled, to reach zero speed.

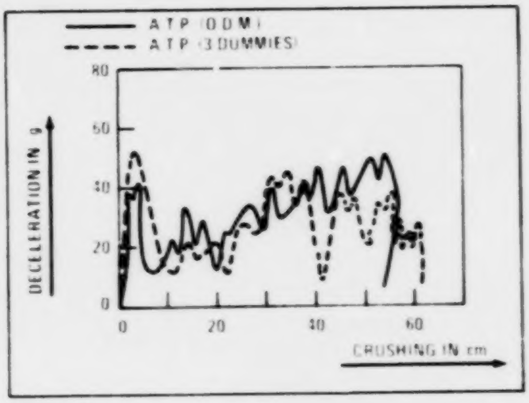


Figure 33

RESTRAINT SYSTEMS

Mathematical Model Simulating Dummy Behavior During A Frontal Impact

ESTABLISHING THE PROBLEM

Optimization in safety belts is a very complex problem, involving a large number of parameters whether considered from the safety point of view, as an absolute, or with regard to existing standards of conformity.

We felt that by directly undertaking a very sophisticated model, we would fail to observe the main phenomenon and the effect of those parameters, which seemed to us, at first sight, the most important.

The method we selected represents, in a first stage, only those parameters we think most important. Furthermore, we had to be quick, a whole philosophy had to be evolved, and particular attention paid to standards.

We have thus prepared two calculation stages simultaneously, these being intended to facilitate our approach to the third stage.

In the first two stages, we only considered dummy movements in a longitudinal plane parallel to the vehicle plane of symmetry.

In the first stage, we used a simple model with three degrees of freedom. This model was processed in an analog computer and enabled us to narrow down the problem of the dummy restrained by belts only when moving forward. This model has been completed, in some cases, by an even simpler model to simplify interpretation of the results obtained.

In the second stage, processed in a 370-165 computer, we used a model with eleven degrees of freedom representing as well as possible an Alderson 50th percentile dummy. This model enabled us, in the first place, to study all the movements of the dummy during impact, including impact-rebound, taking the seat into account. Next, it was possible to study this same movement but, this time, taking into account any obstacles that the dummy could hit within the passenger compartment (steering wheel, dashboard, structure, etc.).

SIMPLIFIED MODEL

The dummy (Figure 34) is reduced to a pelvis, a

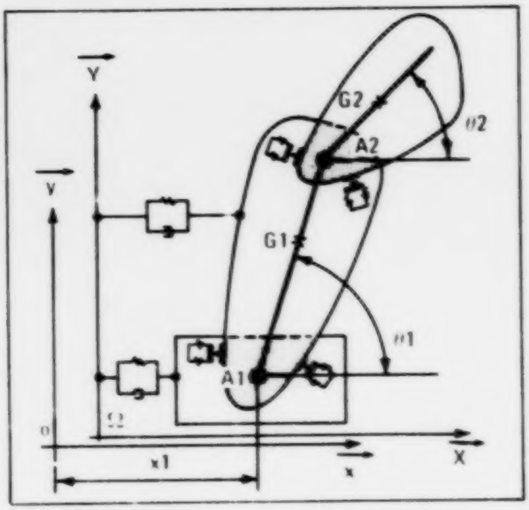


Figure 34

torso and a head. This is the model that D. F. Moore (West Virginia University) presented to the Detroit and Brussels Congresses.

This model has three degrees of freedom; two angles $\theta 1$, $\theta 2$ and 1 displacement x. The belts are horizontal; one is attached to the torso, the other to the pelvis. They possess resiliency (K) and shock absorption (μ). The pelvis slides with friction.

The two articulations comprise a resiliency, a shock absorbing ability, and a friction ability; they are limited by flexible stops. The vehicle horizontal deceleration curve is plotted, and the speed and distance travelled, as a time function, are deduced by integrating. The dummy movement equations are written using the basic dynamic equations and the LaGrange equations to cross check.

Computing Procedure

All parameters were carefully measured or computed. An attempt was made to reproduce an Alderson dummy, which has been used in a certain number of reproductibility tests, as regards mass, inertia and articulation characteristics.

Some characteristics, such as belt rigidity and articulation stiffness, were measured. Others were computed, particularly for the neck, since representation is nonetheless sketchy. After analyzing impact sled testing, an average case, typical of the tests, was selected. The results taken into consideration are as follows:

- Accelerations along x and y (reference system relating to the head)
- Accelerations along x and y (reference system relating to torso)
- Loads through belts
- Deceleration of vehicle (or impact-sled) in as simple a form as possible.

We used this test as a basis. Afterwards, we attempted, by computation, to adapt ourselves to these results by taking a deceleration curve as close as possible to that of the vehicle and manipulating the parameters that we had had most trouble in defining. These are, more particularly, shock absorption, friction in the articulations, and head and torso positions in relation to their stops. We did not touch on inertia and masses or the belts (rigidity curve). We obtained the following results which we have called "basic case" (Figures 35 through 37).

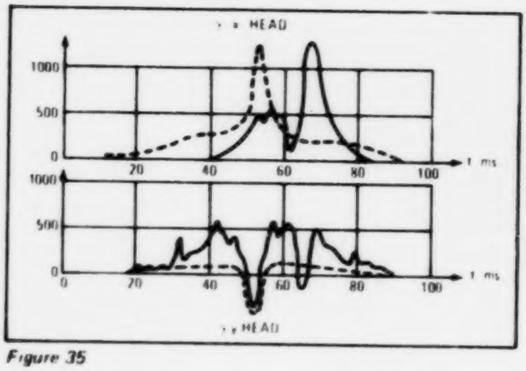


Figure 35

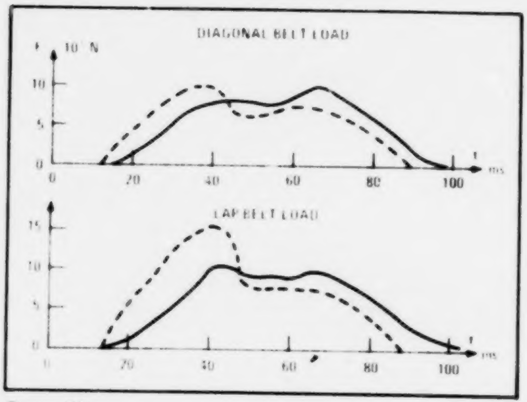


Figure 36

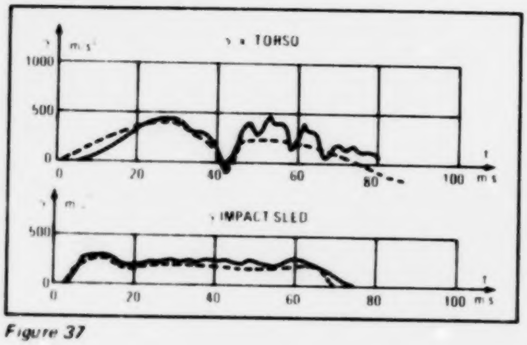


Figure 37

Comments On The Basic Case

We must at first bear in mind, when considering these results, that the model is *very simple* and very approximate; extremely close correlations were thus not to be expected. For example, acceleration γ x for the head constitutes only one point in computations. In fact, we only used one stop, while the dummy has two stops (base of neck and chin). Accelerations γ y are far smaller in computed form than during testing, although the forms are approximate, this is because the model does not move vertically.

Examples Of Results Obtained

Using a model which we consider suitable for orientating our work, despite its simplicity, enabled us to judge the effect, particularly the qualitative effect, of some parameters starting from the basic case.

- Effect of belt tightness

  In the first stage, we multiplied the belt rigidity by 2, 3, 4, 6, 8 and then divided this same rigidity by 2, 4, 8. Results are shown in Figure 38.

  For reference we also dealt with a case in which we only varied the tightness of the diagonal belt (Figure 39).
- Effect of play (clearance)

  Play is 12 mm in the basic case, we have attributed it with the following values 0, 24, 48 mm. We have also brought out the fact that play effect varied according to the shape of the impact-sled deceleration curve (Figure 40).
- Shock absorption effect
- Deceleration curve effect

  For our work we selected three different deceleration curve shapes: crenellation, a curve with a peak at the beginning of impact and another with the same peak, but at the end of impact.

TIGHTNESS	8	4	2	1	x 2	x 3	[illegible]	[illegible]	[illegible]
γx HEAD [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
γx HEAD [illegible]			[illegible]	[illegible]	[illegible]		[illegible]	[illegible]	[illegible]
γx CHEST [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
DIAGONAL BELT LOAD (da N)		[illegible]	[illegible]	[illegible]	[illegible]		[illegible]	[illegible]	[illegible]
LAP BELT LOAD (da N)		[illegible]	[illegible]	[illegible]	[illegible]		[illegible]	[illegible]	[illegible]

Figure 38

CLEARANCE (mm)	0	12	24	48
γx HEAD (m/s²)	900	1300	1800	2500
γx CHEST (m/s²)	350	400	420	500
DIAGONAL BELT LOAD (da N)	900	1000	1050	1200
LAP BELT LOAD (da N)	1400	1550	1700	2000

Figure 39

DAMPING	0	0.05	0.1	0.2	0.5
γx HEAD (m/s²)	1900	1600	1300	900	600
γx CHEST (m/s²)	500	400	400	350	300
DIAGONAL BELT LOAD (da N)	1200	1100	1000	900	700
LAP BELT LOAD (da N)	1900	1800	1550	1500	1200

Figure 40

TWO-DIMENSIONAL MODEL WITH 11 DEGREES OF FREEDOM

Description Of The Model

This model, which is intended to simulate reality more closely, has seven sections for the body and two for the arms (Figure 41). Mass and inertia are attributed to each section. Some sections which sustain contact have a volume represented by a circle, as in the case of the head, pelvis, knee and foot, or by arcs of a circle as in the case of the torso. These sections are connected by articulations. Each articulation has a constant friction, resiliency with flexible stops, and shock absorption.

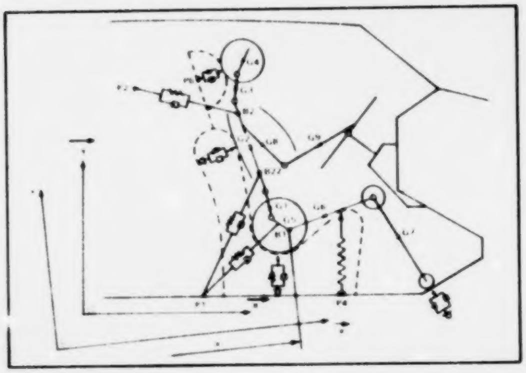


Figure 41

The 3-point belt is represented by three straps; the upper strap simulates the diagonal belt, the other two, the lap belt and the lower portion of the diagonal belt. Each belt has resiliency and shock absorption. The head rest, seat back, rear portion of the seat and the foot rest on the flooring have resiliency and shock absorption. Contact is by friction.

The forward portion of the seat has resiliency only; contact is non-friction and that portion is fixed, like the head rest, while the foot, rear portion of the seat, and seat back, are mobile supports.

All contacts with the passenger compartment are considered as mobile supports with resiliency and shock absorption. Movement is horizontal. Vehicle linear and angular decelerations, as well as steering wheel rearward displacement, are entered.

Computation Process

The process developed was the same as for the simplified model. Simplified model results have been employed to develop this program.

Results

To date, a certain number of results, which are difficult to present simply, have been obtained. The model is under experimentation.

THREE-DIMENSION MODEL WITH 40 DEGREES OF FREEDOM

This model, which has the advantage of representing a dummy's movement during frontal impact with a greater accuracy, will also enable it to be studied during oblique impact. It will shortly be in use as a result of our collaboration with Calspan.

PRACTICAL TESTING

We began our analysis of restraint systems the same way we did structure research work and improvement, by a detailed examination of safety belts fitted as standard equipment for our vehicles. Our aim was to go further than the certification tests, required by regulations presently in force.

The testing sequence will be defined, taking into account computation results obtained with the simplified model.

To summarize these results, we varied the main parameters: belt stiffness, initial play, shock absorption, vehicle deceleration.

The most important parameter turns out to be shock absorption, for head or torso γ x accelerations. If belt stiffness is decreased, dummy movement is increased, which is not compatible with a smaller vehicle. By increasing stiffness, loads on the belt straps increase considerably. We are afraid of the consequences on the human body.

The manufacturer cannot govern initial play on the belt since he must necessarily allow the user to adjust it. Reel or return systems do not prevent the user from being in an unfavorable position at the moment of impact.

We have already discussed the deceleration curves (Figure 27, for example). Nonetheless, one comment must be made. If we took the deceleration curve showing a peak at the end of impact as the ideal curve to be obtained, we would run the risk of obtaining this configuration only for a very restricted speed range.

"Standard" Belt Testing

These tests, together with the following ones, were carried out by means of a spring impact-sled (Figure 42). Sled acceleration, at the moment of release, is less than 3 g. Sled deceleration, simulating vehicle crushing, was obtained by logitudinal crushing of metal tubes.

The impact-sled is very rigid and has points for anchoring the belts and is fitted with a seat similar to that of a vehicle. Nonetheless, the seat and seat back have been considerably stiffened compared to a standard seat.

Measurements Taken

Velocity — Electronic Hewlett Packard chronometer, type 5216 and photoelectric barrier.

Acceleration — (Head and thorax) by sets of three Setra sensors, types 100 and 110, triaxially mounted; impact-sled — two Setra sensors, type 110, ± 100 g placed at the center of the sled.

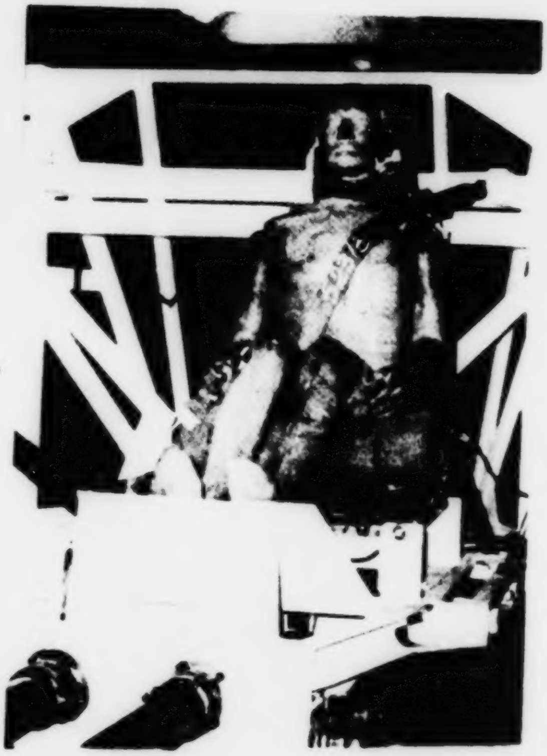


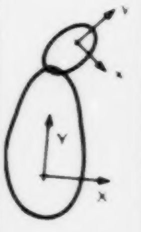
Figure 42

Belt Loads — Lebow type sensors. These various data are collected on a VR 3360 type CEC magnetic recorder, then reproduced and filtered, according to the SAE J211 recommendation, on an EPI 12 type CDC magnetic recorder.

Filming — Adjusting points are placed on the dummy and on the impact-sled, and the test is laterally filmed by means of a Hycam camera at a speed of 500 frames per second. These tests enabled us to adapt our testing method and supply our computing department with coefficients. We used Alderson VIP 50 A dummies, and then the Sierra 292 - 1050 type. We selected the latter type of dummy for our further work. We made use of the table in Figure 43 as a reference basis. Values selected are the average values for five tests.

EFFECT OF PLAY

We checked this configuration on a borderline case, i.e. by initially applying a tensile stress to the belt straps. This load is approximately 250 daN - 300 daN (Figure 44). We are presently working on a



REFERENCE BASIS					
ACCELERATION IN g					
HEAD			TORSO		
	+	-		+	-
X	132	156	X	53	20
Y	71	16	Y	26	19

Figure 43

system for taking up the play in belts at the moment of impact. The device in question is a pyrotechnical action jack.

SHOCK ABSORPTION EFFECT

Computation emphasizes the value of a high degree of belt shock absorption, even if we only consider the case of forward movement. We have designed a belt capable of progressive elongation at a constant load (Figure 45).

CLEARANCE INFLUENCE					
ACCELERATION IN g					
HEAD			TORSO		
	+			+	
X	117	77	X	47	76
Y	62		Y	24	17

Figure 44

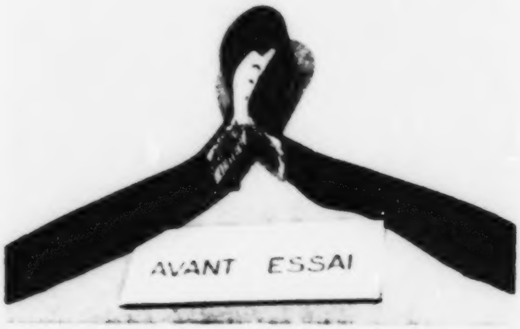


Figure 45

This belt is chiefly constituted by a 4% rigidity strap having a breaking load of 2,600 daN - 2,800 daN. Five differential elements are sewn on the diagonal belt, and are included in a 280 mm long loop of the belt.

Before producing the belts, which were tested with a dummy (Figure 46) by means of an impact-sled, we carried out a check by means of a static test (Figure 47).

With an impact velocity of 50 km/h (14 m/s), we obtained the results shown in Figures 48-51.

Note the five strap breakings and the similarity to the static test curve (Figure 48).

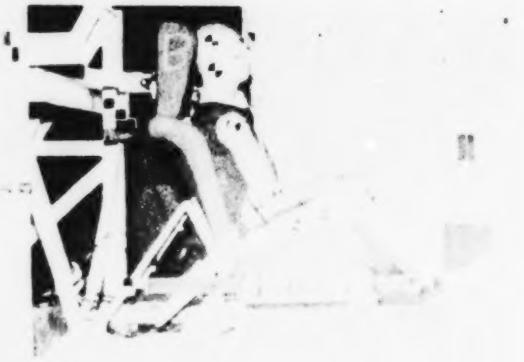


Figure 46

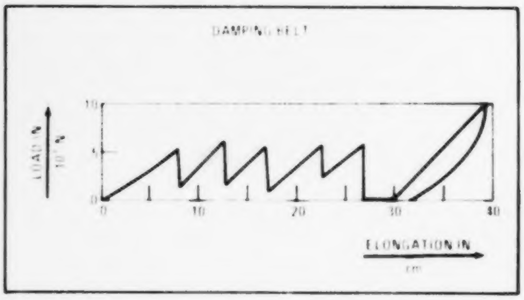


Figure 47

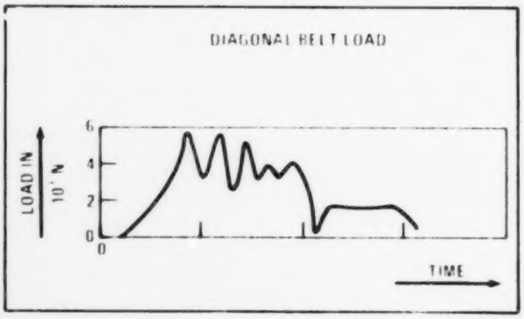
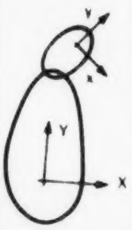


Figure 48



DAMPING INFLUENCE ACCELERATION IN g					
HEAD			TORSO		
	+	-		+	-
x_1	35	10	X_1	50	20
y_1	42	1	Y_1	14	12

Figure 49

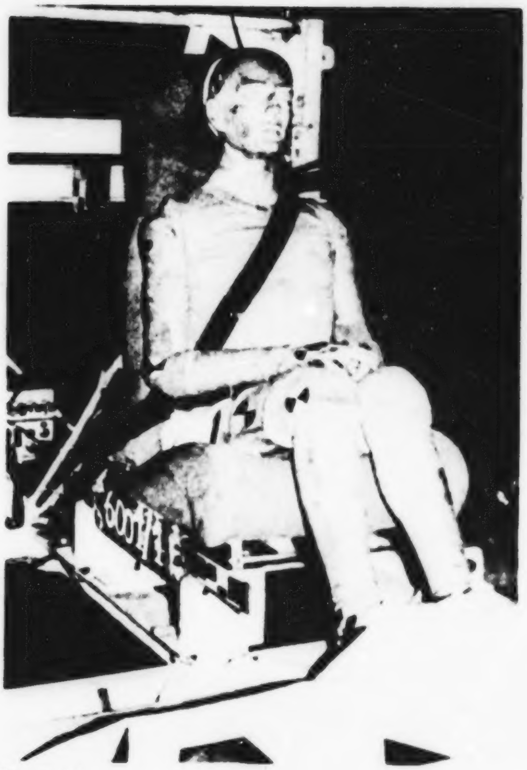


Figure 50

TEST ON DYANE ATP VEHICLE

The dummies placed in the ATP vehicle for the 56 km/h test were restrained by belts of the internal shock absorption type (Figure 52). After testing, we noted the deteriorations to the belt and, more particularly, the breaking of differential elements, ascertaining the importance of their stresses; we based our conclusions on the test films.

Figure 51

Figure 52

Driver's Belt

Four strands were broken (Figure 53). Torso and head displacements were extremely important since the torso of the driver dummy hit the rim of the steering wheel, and its head hit the hub.

Front Passenger's Belt

All five strands were broken (Figure 54). The dummy must have therefore undergone a greater load than the driver. It can be concluded, from examination after testing, and analysis of the film, that this was due to the rear passenger "submarining," its knees bearing heavily upon the small of the front passenger's back (Figure 55).

Rear Passenger's Belt

Only one strand broke (Figure 56), but the dummy was subjected to an important movement "submarining." This movement should be attributed to the flexibility of the seat.

This interference between dummies is difficult to deal with using the impact-sled. We must, however, make allowance for this. It should be noted that this problem is accentuated in a small vehicle, in which the space between the front and rear occupants is necessarily restricted.

We are proceeding with our research work by including, both in computations and in impact-sled testing, the external elements (structure), and a greater seat flexibility; this being in the 50 to 65 km/h speed range.

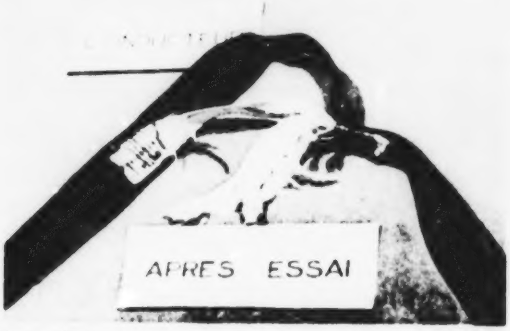


Figure 53

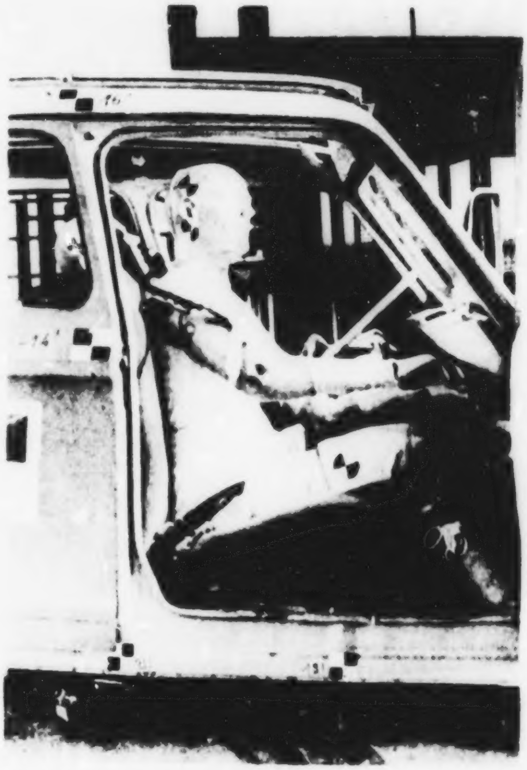


Figure 55

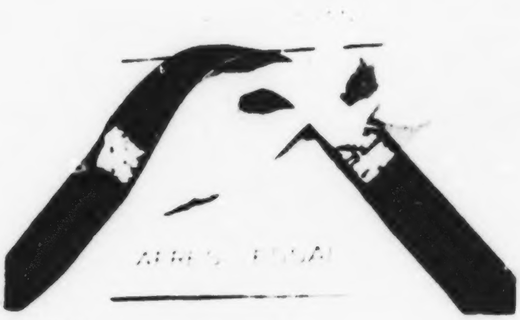


Figure 54

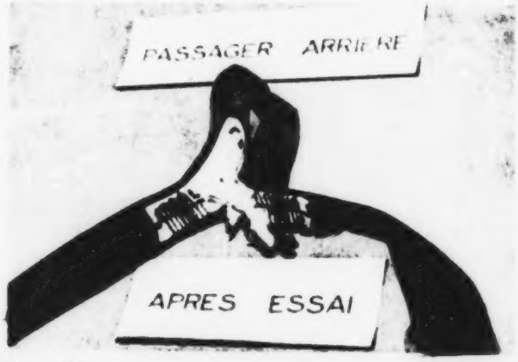


Figure 56

MAIN COMMENTS

Structure At Curbweight

Examination of characteristic curves (Figure 27) for the ATP Dyane and the Standard Dyane shows a rapid increase in structure average deceleration rate when the crushing distance decreases. With a 65 cm crushing, we reach a threshold beyond which the residual volume of the passenger compartment is concerned.

Structure At Laden Weight

The presence of dummies (Figure 33) increases time and crushing distance and tends to decrease, after a 30 cm crushing, i.e. approximately 50% of total crushing, the deceleration rate for the whole of the structure.

Effects On Restraint Systems

In order to improve the efficiency of these systems, we can vary stiffness, initial play, and shock absorption.

Stiffness cannot exceed the tolerable limit for the thorax and for the pelvic zone. Figure 38 gives an idea of the increase of loads caused by increased belt stiffness.

Play research has been carried out to eliminate play between belt and occupant, at the moment of impact.

For shock absorption we are trying to obtain a progressive elongation for a constant load.

CONSEQUENCES

The problem of a sub-subcompact car is very simply illustrated in Figure 57. It is summed up with respect to secondary safety and frontal impact, in particular, by a distance between H point and the front bumper.

If, as we desire for economic reasons and traffic conditions, particularly urban conditions, this distance should decrease by 24 cm in relation to our Dyane based ATP research (i.e. give a crushing distance: 41cm) we can say that the vehicle deceleration reaches 80 g with an impact velocity of 65 km/h (Figure 58).

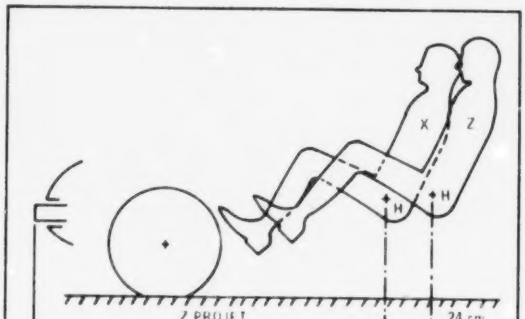


Figure 57

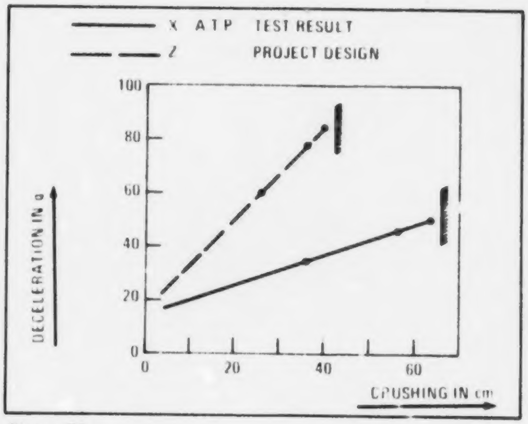


Figure 58

Two questions are raised with this deceleration rate — What happens to the restraint systems? What are the effects on human physiological tolerance.

In fact, what effect is there as regards sub-subcompact vehicle safety? We would refer to the Onser inquiry concerning 14 French departments for the six month period June to November 1969, which demonstrates that the death rate for front occupants is *significantly lower for some types of sub-subcompact vehicles, including Citroen 2 CV and Dyane*.

CONCLUSION

Before giving an opinion on how effectively the estimated performance of the vehicle structure protects occupants during collisions, it is essential to be acquainted with human tolerance, with respect to load and deceleration. Many surveys are being carried out at present and, in this field, accident inquiries followed by possible reproducible tests are essential. Our next research stage must not be limited to structural improvements and improvements in equipment, but, indeed, must take into account what happens in reality to human beings.

PERPENDICULAR SIDE-ON COLLISION BETWEEN TWO VEHICLES

Structure Improvement — Criticisms Of The Standard Structure

During a side-on collision test at 32 km/h involving a standard GS vehicle with a total weight of 944 kg (without dummy) and a standard Dyane vehicle with a total weight of 796 kg (with two dummies), we observed the following damage to the Dyane (target vehicle) (Figures 59-61): the side doors (impact side) buckled and jammed and the lower front hinge pulled down; the body side member crushed; and welding points to the side flooring untacked. The maximum accelerations of the vehicles were: striking vehicle GS = - 8 g; target vehicle: Dyane = 13 g.

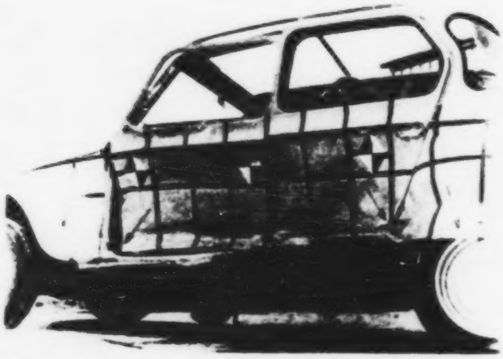


Figure 59

Figure 60

Figure 61

Design Improvement Research

- increase in hollow body for side members (Figure 62).
- add two cross members under front seats
- center pillar-base reinforcement

GS Against Dyane ATP

This side-on collision test at 33 km/h was conducted with a standard GS vehicle (stripped of trim, without dummy) and weighing 928 kg, and an ATP Dyane vehicle (stripped of its mechanical components, with two dummies) weighing 772 kg (Figure 63).

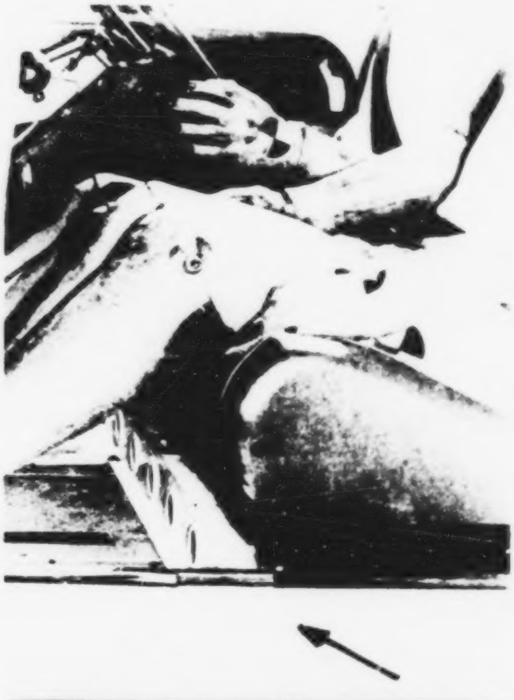


Figure 62

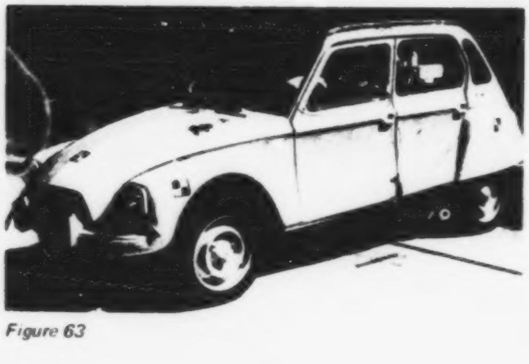


Figure 63

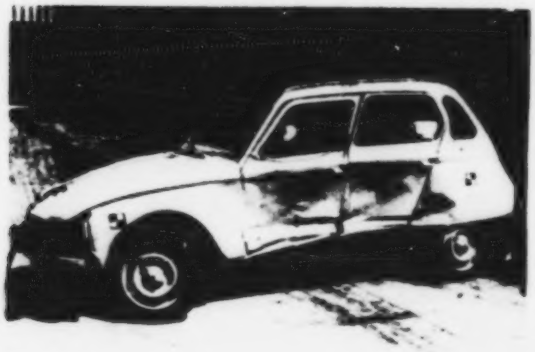


Figure 64

ANALYSIS OF RESULTS

We noted the following damage to the ATP Dyane (target vehicle) (Figures 64 and 65): the side doors (impact side) buckled but were capable of opening; the center pillar and body side member buckled and the side flooring crumpled. The maximum accelerations of the vehicles were: striking vehicle: GS = -10 g; target vehicle: Dyane ATP = 13 g.

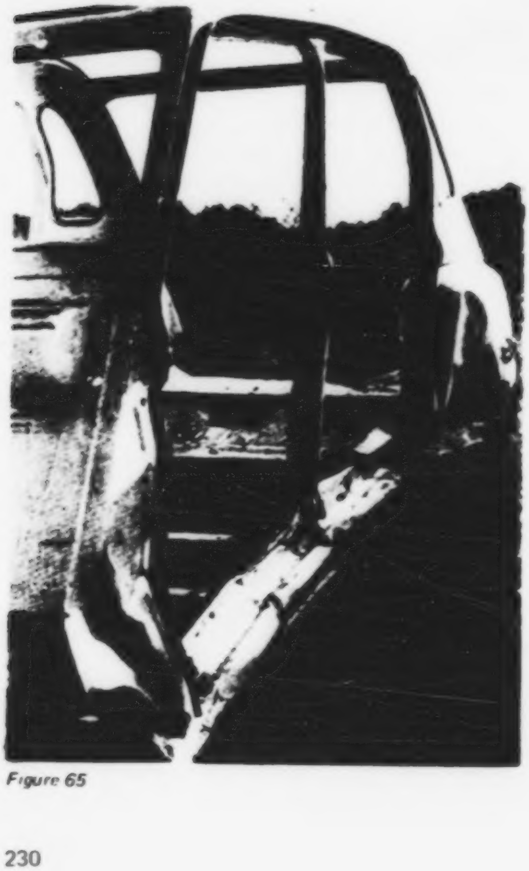


Figure 65

CONCLUSION

Thanks to reinforcements, damage was less extensive than with the standard vehicle (Compare Figures 61 and 65). There was less collapse to the underframe; the side panel-side flooring coupling was not torn away; and the front seat was very slightly buckled, hence there was improved residual space.

APPENDIX – RESEARCH METHOD USING SUCCESSIVE DROPS TO SIMULATE A SINGLE COLLISION AT A HIGHER SPEED

Acceleration Curve As A Function Of Cumulative Crushing

To obtain an acceleration curve as a function of crushing in collision tests, vehicle passenger compartment decelerations are recorded as a time function. By doubly integrating these accelerations (Figure 66), crushing is obtained at the front of the passenger compartment as a time function. (Note that the same result can be obtained by digitally stripping the film of impact). From these two data sets we can plot acceleration values (or load value, when mass is known) as a function of crushing.

To adjust acceleration-crushing curves, assume the hypothesis that, for a sheet metal assembly, buckling loads do not depend on velocity (a hypothesis which has been approximately checked in several tests). In each test, the passenger compartment velocity decreases and then reaches 0; crushing x_1 is then at a maximum and we have a load F_1 at this moment, i.e. $F_1 = M\gamma$.

Velocity is then reversed and the spring rebound corresponds to a decrease in crushing.

If the same vehicle is subjected to a second impact, the beginning crushing corresponds with the spring rebound travel at the end of the first impact, until load F_1 is reached, and then the plastic deflection resumes.

Therefore, we join the acceleration-crushing curves for the two tests at this point. An interesting cross check can be made by using the permanent crushing value. This crushing value, recorded after the first test, must coincide with the origin of the following impact curve.

ACCELERATION CURVE, CUMULATIVE AS A FUNCTION OF TIME

Usefulness Of This Type Of Curve

The acceleration curve, as a function of crushing (Figure 67), allows us to characterize a structure, but as regards dummy kinematic research, it is necessary to refer to the acceleration curve as a function of

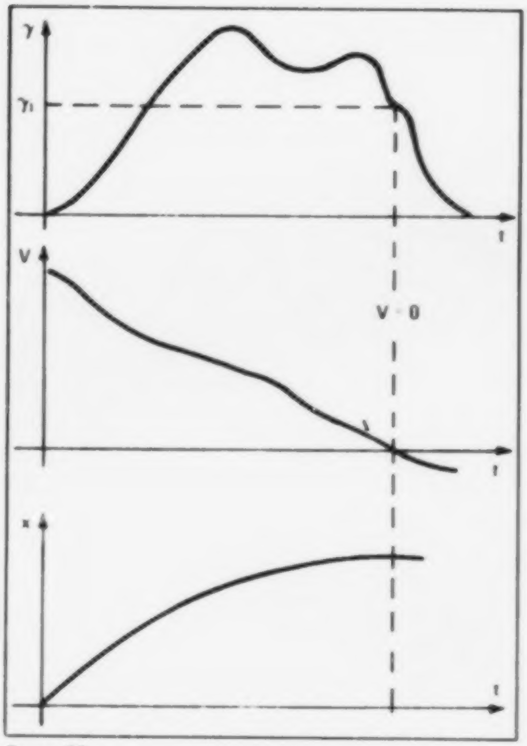


Figure 66

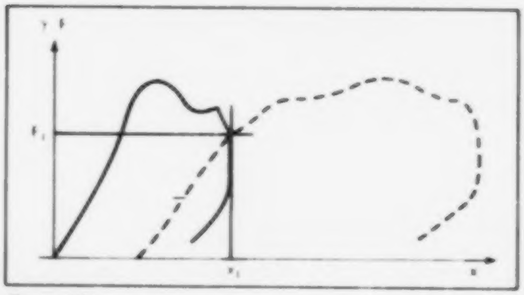


Figure 67

time. The vehicle acts upon the dummy by decelerations transmitted to it during the course of time, through seat belts, air bags or any obstacle.

Reconstituting A Single Curve On The Basis Of Successive Drops (Figure 68).

We still assume the hypothesis that the loads, and hence decelerations, are not dependent on crushing velocity at each moment. It is simply necessary to reconstitute a time scale corresponding to an initial velocity greater than test velocity. For instance, one can reconstitute an impact at 65 km/h, (height of drop is 16.7 m), with the following three drops: 2.40 m (24.5 km/h), 7.05 m (42.3 km/h) and

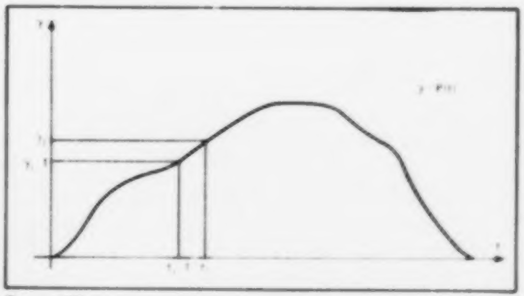


Figure 68

7.25 m (42.8 km/h). The last drop will be used without any change, but the initial velocity of the first drop will be considered equal to 65 km/h, instead of 24.5 km/h; the initial velocity of the second drop corresponds to residual height = 7.05 m + 7.25 m, i.e. 60 km/h.

Computation Process

For the first drop: initial test velocity:
v_0 = 24.5 km/h; simulated initial velocity
V_0 = 65 km/h
Digitalizing the real curve obtained $\gamma = f(t)$ with a sufficiently small time interval

$$\text{Reading pairs:} \begin{cases} \gamma_i \\ t_i \end{cases}$$

Computing actual velocity as a function of:

$$dv = \gamma\, dt$$

$$v = \int_0^t \gamma\, dt \text{ or } v_i = v_{i-1} + \frac{1}{2}(\gamma_i + \gamma_{i-1})\, dt$$

$$\text{Hence we obtain the pairs:} \begin{cases} \gamma_i \\ \gamma_i \end{cases}$$

In order to compute simulated velocity as a function of γ, let us write that for the crushing of a given section, the energy transformed is independent of crushing velocity.

$$d\left(\frac{1}{2}\, m\, v^2\right) = d\left(\frac{1}{2}\, m\, V^2\right)$$

either $d(v^2) = d(V^2)$ or $V_i^2 - V_0^2 = v_i^2 - v_0^2$

$$\text{i.e.: } V_i = \sqrt{v_i^2 + (V_0^2 - v_0^2)}$$

$$\text{we obtain the pairs:} \begin{cases} \gamma_i \\ V_i \end{cases}$$

To obtain the new time values:

$$dV = \gamma dT \text{ i.e. } T_i = T_{(i-1)} + 2 \frac{V_i - V_{i-1}}{\gamma_i + \gamma_{i-1}}$$

we therefore know the pairs: $\left\{ \begin{array}{l} \gamma_i \\ T_i \end{array} \right.$

which enable the transformed curve to be plotted (T) $\gamma = f(T)$

It simply remains to juxtapose the consecutive curves by joining them up to the point corresponding to zero speed in the actual test, by means of the method used for the curves - acceleration - crushing.

CONCLUSION

Our research in the second stage of the ATP investigation revealed that the survival problem of passengers restrained by an active restraint system is established during a perpendicular fixed barrier frontal impact at a speed exceeding 55 km/h.

We confirm that a rulemaking action prescribing high impact performances for all types of vehicles, whatever may be their mass and dimension, condemns the small, sub-subcompact, European vehicle.

Complying only with the criterion which implies maintaining a residual space after the impact produces small car occupant deceleration levels such that an adequate protective system remains to be invented; any adequate system is likely to be so burdensome as to tend to condemn the sub-subcompact vehicle.

We request that prescribed performances make allowance for facts. These appear, through the accident inquiries from which, at present, European manufacturers, engaged in this action, may draw the first conclusions.

THE UNITED KINGDOM TECHNICAL PRESENTATION

SECTION TWO PART 4

INTRODUCTION

H. TAYLOR, *Head of Safety Department*
Transport and Road Research Laboratory

It is my pleasure to introduce the United Kingdom presentations. It is not our intention to cover all of the British car safety programs in detail, because it would be impossible in the time available. Instead, we have selected those items which we believe to be of major importance at this time. We shall begin with a paper by Mr. R. D. Lister of the Transport and Road Research Laboratory, who will describe the United Kingdom requirements for a practical safety car.

SECTION TWO

PART 4

UNITED KINGDOM REQUIREMENTS FOR A PRACTICAL SAFETY CAR

DUNCAN LISTER, *Head of Vehicles Division*
Safety Department
Transport and Road Research Laboratory

You will have heard from the *United Kingdom Status Report* given on Tuesday that we are now moving into Phase 2 of our Car Safety Programme. In this phase we will be able to exploit the projects developed under our Phase 1 programme and bring them together to form the basis of a practical safety car capable of being produced with normal mass production methods.

This also provides an opportunity to look at some of the lessons learned from the experimental safety vehicle in which the main emphasis was car occupant protection. Considerable ingenuity and expertise was exercised, and it was demonstrated that a car occupant could be afforded a high degree of protection, particularly in frontal collisions; however, not all road casualties are car occupants, far from it, and there is a very serious constraint in the overall reduction of road casualties, if too much emphasis is placed on car occupant protection only.

It is clear from pervious ESV conferences that there is a conflict of requirements in many of the features affecting safety, e.g., attention was drawn to the need for the design of cars to reduce pedestrian injuries and taking account of this could mitigate, to some extent, the protection of car occupants. Also, the need for vehicles to be compatible in front and side impacts is now well recognized, which again may mean that the occupant protection in frontal impacts is affected. Furthermore, because of the greater benefits that can accrue from preventing accidents, greater effort should be spent on primary safety.

In considering Phase 2 of the Car Safety Programme, we have tried to exercise some degree of judgment in deciding which particular safety measures to promote. This has been done on the basis of our own accident statistics; firstly, the overall national data in which all the injury-producing accidents are recorded; this covers some 350,000 casualties annually. In addition, TRRL and the Birmingham University team have their own detailed accident injury investigations in which very fine detail of the accident, the vehicle damage, and the injuries sustained, are all recorded. From these considerations, the United Kingdom requirement for a practical safety car has been drawn up and is given in Appendix 1. These requirements presuppose that all current UK "Construction and Use Regulations" and EEC legislation will be met and that due account is taken of anticipated EEC legislation. Emphasis has been placed on the aspects of safety given in Figure 1.

PRIMARY SAFETY

In the field of primary safety, instability under braking conditions and under varying road surface conditions is a significant factor in road accidents. The requirements to be met are designed to ensure that maximum braking performance and stability is maintained at all times. These are most likely to be met by the use of an advanced anti-lock brake system, though other methods are not excluded.

In the case of handling requirements, we are not convinced that a proper relationship has been established between a measure of handling performance and the risk of accident involvement. There is much work still to be done in this field and the United Kingdom is actively pursuing this aspect of its accident investigation work. In the meantime, it is reasonable to demand that in the interests of safety, handling characteristics should not change when the car is used at its extreme range of load, tire pressure, tire wear and be stable in side winds, as well as retaining directional stability under differential braking forces.

Good progress on driving aids and driver information has been made, and it is our intention that these should be incorporated in our practical safety car and that provision should be made for further aids that may come along in the future. Many of these may need to be considered in conjunction with road features and road design, and draws attention to the wider aspects on which safety must be pursued. In

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general, these driving aids have the objective of modifying driver behavioral patterns and as it has been shown that driver behavior is a prime factor in accident causation, there are good prospects of large "pay-offs" in primary safety.

SECONDARY SAFETY

As indicated at the Third ESV Conference at Washington, we aim to base our requirements on performance, rather than on design, and in the field of secondary safety this is particularly so in the case of occupant protection. Because the special requirements for a restrained occupant and an unrestrained occupant conflict, a choice has to be made as to which condition should receive the main consideration. We have decided that the main occupant protection should come from some form of restraint, and that occupants could be expected, or even compelled, to use such devices and the maximum protection should be afforded in such cases. It is, however, appreciated that there may be some instances where restraint is not possible, e.g., on medical grounds, and also to act as a back-up system in the event of failure of the restraint, so that at least some minimum level of protection in the form of padding material or deformable structures should be available in these cases, and a lower degree of protection would be given.

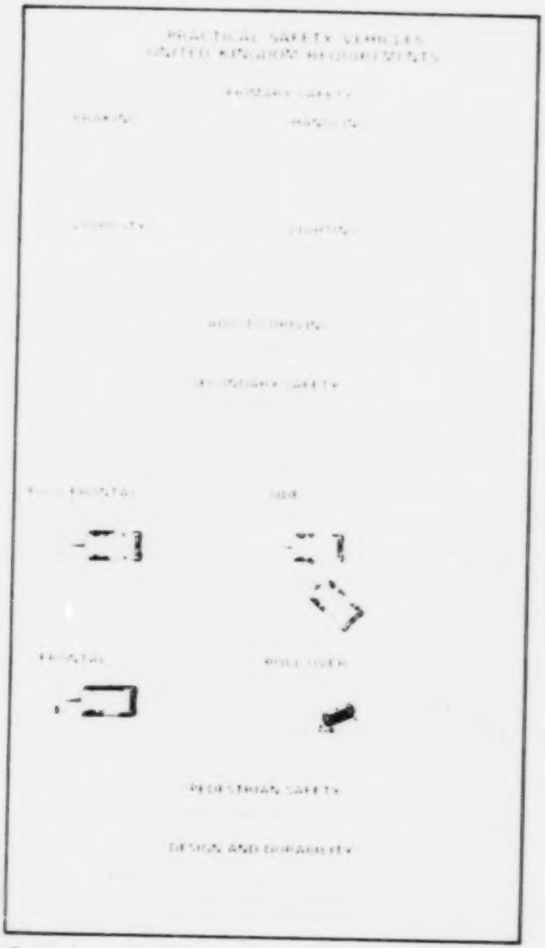

Figure 1

In our requirements, tests are given for frontal and side impacts and also rollover. We have estimated that the maximum frontal impact test condition at 60 km/h will cover at least 80% of the casualties occurring in frontal impacts and that a considerable reduction of injury risk to the remainder would follow. As the proportion of protection required to be provided by the interior structure is determined by the full frontal impact test with unrestrained, as well as restrained occupant dummies, the remaining impact and rollover tests are to be performed with restrained occupant dummies only. Furthermore, the quarter frontal impact, which may be carried out on either side of the car, is much more typical of real life accident situations, and is a case which must be included. It is also intended that this test will encourage manufacturers to design front structures which lead to glancing, rather than interlocking, impacts.

A side impact test is called for with the intention of ensuring that there is an adequate energy-absorbing structure on the inside of the vehicle. It is also required to show that intrusion has been controlled to reduce the risk of crushing-type injuries. This test will promote designs of side structures and front structures of striking cars which are matched, and will also encourage designs which convert side impacts to glancing impacts. These objectives are consistent and compatible with the quarter frontal impact test.

Specially developed test dummies, developed by the Transport and Road Research Laboratory for side impacts, with means for measuring transverse loads at selected key points, will be used to evaluate the level of protection provided in the struck-car-in-side impact tests.

For the frontal tests, assessment of the protection afforded is by use of an occupant testing device and it is necessary that the performance and design characteristics should be closely specified. The development of this dummy test device is the subject of another United Kingdom project under the ESV Programme and will be dealt with by a later paper in this meeting.

PEDESTRIAN SAFETY

In addition to the requirements for the protection of vehicle occupants, our practical safety car is required to provide for pedestrian protection. At the moment, this has been expressed in terms of dimensional requirements for the front of the vehicle structure and in impact tests for the vehicle bumper structure. These are intended to minimize injuries to the leg and to control the trajectory of the pedestrian when struck by a vehicle, so as to minimize the risk of being thrown on the road by the impact.

Some details of development work at the Transport and Road Research Laboratory is given in Appendix 2. It is clear that some means of retaining a pedestrian on this vehicle until it is brought to rest is desirable.

GENERAL

It is intended that during the development of the practical safety car, requirements will be updated in the light of test data generated during the work, and also to ensure that all safety features which can be incorporated at an economic cost relative to their prospects for improving safety, will be included.

In addition to the requirements for safety, the car shall have a reasonable life expectancy in terms of durability and ease of maintenance. So that, at the end of this development, we shall have the solution to a practical safety car capable of being produced economically and in large quantities which is acceptable to the public.

APPENDIX I
SAFETY CAR REQUIREMENTS

INTRODUCTION

In general, the safety car shall meet current "Construction and Use" and EEC requirements, unless these conflict with anything expressly stated in this specification or unless discarded by mutual agreement.

SAFETY CAR DESIGN BASIS

Secondary Safety – Occupant Protection

The levels of occupant protection chosen for this specification cover impact severities, which account for most of the injuries to car occupants in the United Kingdom. However, if design work has already shown that greater protection can readily be provided, then this extra protection is required.

The choice of restraint system is critical to the occupant protection design. In order to correspond to the most likely situation in the UK in a few years' time, it is required that a belt system be used that incorporates features which provide maximum comfort and convenience, together with any energy-absorption system necessary for providing maximum occupant protection. To ensure that belts are not worn loosely, it is required that inertia reel belts be used in front seat positions. It should be borne in mind that it may be required to fit passive belts to these vehicles in the future. To cover exemptions and exceptional cases, tests are to be carried out with unrestrained dummies in all seating positions.

Though only full-vehicle tests with complete dummies are specified, it is intended that the vehicle manufacturer should satisfy himself that the design would pass similar tests at different angles and dummy attitudes and sizes, possibly by carrying out subsidiary tests on parts of the vehicle interior structure using head, chest and knee forms.

The purpose of the tests detailed below with full vehicles and complete dummies is as follows:

- The head-on barrier test at 60 km/h is to check the maximum frontal impact condition thought to be necessary for 80 percent of casualties in frontal impacts in the United Kingdom. The quarter partial head-on barrier tests at 50 km/h is partially to represent the large number of frontal impacts of this type, but it is also intended to encourage the development of front structures, which lead to glancing, rather than locking, impacts. No provision has been made for an angled frontal test impact because though similar impacts occur in accidents, protection is catered for by the subsidiary tests mentioned above.
- The side-impact test is intended to demonstrate that there is an adequate, energy-absorbing structure between the dummy nearest to the impacted side and the impacting object. It is also intended to demonstrate that intrusion has been controlled to reduce the risk of crushing-type injuries and to encourage the design of side structures, which lead to glancing, rather than penetrating impacts. A particular objective is to demonstrate the compatibility of the front corner structure when striking the side of a car.
- No single test of structural resistance to overturning damage to the roof is entirely satisfactory, but the one chosen provides an indication that the roof support structure is adequate in one set of overturning conditions.
- No rear impact is included, because accident studies show that, at present, serious occupant injury following impacts into the rears of cars is

comparatively rare in the United Kingdom. The best of current standards of rear structural design will be followed with due regard being given to the risk of fuel leakage, following petrol tank displacement if it is placed in the rear. Head rests yielding at not less than 900N, when struck by a head, shall be fitted to all seating positions.

IMPACT TESTS TO DEMONSTRATE LEVEL OF OCCUPANT PROTECTION

Test Devices

Anthropomorphic dummies shall be used as test devices, in order to determine the level of protection afforded by the vehicle under test. The Ogle test dummy, developed as a result of the TRRL contract, shall be used for the frontal impact tests. For the car-front to car-side impact the TRRL side impact dummies shall be used and they shall be placed in the seat adjacent to the point of impact and the seat behind it. 50th percentile male dummies shall be used for the tests.

INSTRUMENTATION

All load and deceleration measuring channels shall be capable of recording from DC to 1,000 Hertz for head accelerations, 600 Hertz for femur force and 180 Hertz for other recordings in accordance with SAE J211A. All recordings shall be within ± 5% of the appropriate levels specified below. Displacement measurements shall be recorded, as required, to give a dynamic measurement which shall be accurate to within ± 5% of the levels specified below.

TEST CONDITIONS

Installation Of Test Devices

The test dummies shall be placed in a natural-seated attitude, specified by the representative of TRRL at the demonstration impact tests; the seating positions being those appropriate to the particular test. Seats or controls shall be set up to suit them in the way in which they are likely to be used by the general public. For example, seat belts shall be adjusted by placing the dummy 25 mm forwards of its natural seated position, adjusting the belt without slackness around it and then moving the dummy back into the natural position and allowing any inertia reels fitted to take up any slackness in the straps to which they are attached.

Preparation Of Vehicles

All doors and windows (except where required for filming) shall be closed, but doors shall not be locked. The fuel tank shall be three quarters full of fuel or a liquid of equivalent specific gravity. Tires shall be at the pressures recommended for normal use. The vehicle shall not be constrained during the impact or rollover tests.

TEST METHODS

Full Frontal Barrier Impact

The car containing the unrestrained test devices, as specified above, in all the manufacturer's designated seating positions shall be impacted perpendicularly into the face of a rigid impact test barrier, mass at least 100 Mg, at an impact velocity of 25 ± 2 km/h. The car containing restrained test devices, as specified above, in the manufacturer's designated front seating positions shall be impacted perpendicularly into the face of a rigid impact test barrier, mass at least 100 Mg, at an impact velocity of 60 ± 2 km/h. The full width of the front of the vehicle shall contact the barrier face. The requirements for the fuel system and tolerance levels below shall be met.

Partial Frontal Barrier Impact

The car shall be impacted perpendicularly into the face of a rigid impact test barrier, as described above, but at a speed of 50 ± 2 km/h with restrained test devices in the front seat positions. The driver's side of the front of the car shall overlap the front face of the barrier, so that 25 ± 3 percent of the front of the car is impacted. The vertical edge of the barrier impacted in the test shall be rounded to a radius of 100 mm. The requirements for the fuel system and tolerance levels shall be met.

Side Impact

The car being impacted in the side shall contain the test dummies specially developed for side impacts by TRRL, only in the designated seat positions adjacent to the side being impacted. The brakes of neither car, nor other means of constraint, shall be used until the cars have completed their impact into each other. The surface on which the test is conducted shall be a normal road surface, rigid, level, dry and free from loose material.

Prior to impact, the striking car shall be travelling at 45° to the longitudinal axis of the car being tested from the forward direction at a speed of 30 ± 2 km/h. The line of travel of the side of the impacting car shall pass 30 ± 30 mm behind the nearest extremity of the A-post of the car. If this point cannot be determined because the vehicle is of unusual construction, then the point of impact shall be chosen by the representative of TRRL with the object of impacting that area which is most likely to intrude upon the front seat occupant. The requirements of the fuel system and tolerance levels shall be met.

Overturning

The test shall be carried out with the car containing dummies, as specified, in all designated seating positions. As for the side impact test, this shall be carried out on a rigid, level, dry surface, free from loose materials. The car shall be slid sideways by placing it on a low trolley with free-rolling wheels set at right angles to the car wheels, or by means giving a similar result. It shall be slid at 50 ± 2km/h into a curbstone let into the surface of the test area so that its top is 150 ± 5 mm above the contact patches of test car tires just before they strike the curb. The curb shall extend the whole length of the car and the test area shall be sufficiently large enough so the car shall remain on it until it comes to rest. The requirements of the fuel system and overturning shall be met.

TEST REQUIREMENTS

Fuel System

During a test impact and for five minutes afterward, no fluid from the fuel tank and fuel system or any corrosive liquid shall spill into the passenger compartment and not more than 250 ml in total of such liquids shall be spilled from any containers in the car. No part of the car or its contents shall ignite. No doors shall open and latches shall remain operable. Apart from the door directly impacted in the side impact, all doors and opening roofs shall be operable and require a force of not more than 300N to open them after the test impact. The test dummies shall be contained within the car during the impact.

Frontal Impacts

The following tolerance levels measured on the test devices shall not be exceeded. Unless otherwise specified, the tolerance levels shall be those listed in TRRL leaflet LF 311.

Head Pending agreement as to the suitability of the mounting of the heads on the Ogle and TRRL side impact dummies, the threshold accelerations quoted in Figure 1 of LF 311 shall be regarded as the tolerance levels for head resultant accelerations. Head injury tolerance levels will be revised periodically according to the state-of-the-art.

Neck The angular requirements in LF 311 for the neck shall be complied with.

Chest and Torso It is required that the fore and aft deceleration, measured at the vertebral column of the test dummy at the level of the fourth interspace, shall not exceed 60 g, except for intervals of not more than three milliseconds, and the chest deflection shall not exceed 65 mm. No restriction is put on the loading placed on the test dummy by a lap belt other than the belt shall not slide up off the pelvis and onto the abdomen.

Thighs The axial force in each femur measured behind the knee shall not exceed 4 kN.

Tibia and Fibula and Feet No structure shall be placed so as to impose a localized loading on the tibia and fibula to restrain the torso. The feet shall not be trapped or forced into an unnatural position in relation to the lower legs of the test dummies by any deformation of the bottom of the passenger compartment or by the positions of the driver's foot controls.

Side Impact

The injury tolerance levels shall not exceed those being established by TRRL.

Overturning

No interior dimension of the passenger compartment shall have been reduced by more than 200 mm as a result of overturning. The dummies shall be contained within the passenger compartment during the overturning.

PEDESTRIAN PROTECTION

First point of impact to the pedestrian to be positioned low down, not more than 375 mm above the ground. The front bumper impact characteristics are to be such that fracture of lower limbs of pedestrians struck by the bumper at speeds of up to 15 km/h is unlikely to occur.

Front compartment shape in profile and plan to reduce the chance of a child pedestrian being thrown on to the road ahead of the car. Present indications are that the top edge of the front of the car be not more than about 585 mm above ground and that, preferably, the windscreen is more than 1,400 mm behind this point. Impact specification for the outside surfaces of the top of the front compartment, the windscreen, the A post and the front header.

General requirements for the removal of external projections and placing them within the profile of the body and the impact specification of inevitably protruding objects such as the external mirrors.

BRAKING

As a demonstration safety vehicle, it is intended that the braking system of the car shall be the best that is reasonably available. Good performance on dry surfaces is required with pedal pressures that can be exerted by the general driving population. Wheel anti-locking equipment will probably be required to meet the braking requirements on a curve, on a low skid-resistance surface, on a split-coefficient surface, and for braking during a lane change. A single hydraulic failure should leave the car with adequate braking for normal purposes and the remaining braking should be such that instability is unlikely to occur. Warning at the facia is required for notifying the driver of a major fault in the brake system or excessive wear. The brake system life is planned for half a car's normal life and replacement of the critical parts of the system should be easy and cheap to carry out. To aid checking the condition of the brake system at more frequent intervals, inspection of friction materials should be readily carried out, together with inspection of brake fluid levels. Automatic adjustment of the brakes to take up friction material wear should be included.

TESTS AND REQUIREMENTS FOR THE BRAKING SYSTEM

Braking acceptance tests shall be carried out at the Transport and Road Research Laboratory track on the surfaces specified below.

Cars shall be fitted with tires approved by the vehicle manufacturers and inflated to their recommendations. For tests other than on dry surfaces, the minimum tread depth shall be not less than 75 percent of that when new. Additionally, the controllability and stability of the cars on wet surfaces shall be reported when either the two front or the two rear tires are replaced by similar tires with one mm tread remaining on them. No performance standards are laid down for tests with tires of one mm tread on the cars. Tests for braking distance and overall deceleration measurement shall not require a pedal force in excess of 500N. Tests for the handling and stability of the cars shall be carried out with pedal forces up to 900N applied as rapidly as possible for each test.

All tests to be carried out at normal load and full load as per the following conditions.

1) Normal load — driver and one front passenger
2) Full load

Small car: driver, front passenger, two rear passengers and 50 kg load in boot.

Medium car: driver, front passenger, three rear passengers and 75 kg load in boot.

Larger car: driver, front passenger, three rear passengers and 100 kg load in boot.

During brake applications the transmission shall remain engaged in the gear appropriate to the initial speed involved and shall not be disengaged until the speed is below 8 km/h. Automatic transmission shall be in "Drive" range. Both the foot brake and the hand brake shall be progressive in their application.

BRAKE PERFORMANCE AND CAR STRAIGHT LINE BRAKING STABILITY

Full Braking

The car shall be braked in a straight line on the dry Surface One and any deviation shall be contained within a 3.6 m wide lane with the steering wheel held in the straight ahead position. Overall braking decelerations shall be measured for stops from 15, 30, 45, 60, 75, 90, 100 and 110 km/h to rest and shall exceed 0.8 g. The runs shall be repeated on this surface when dry and on surface two when wet, both tests with the higher pedal force specified above. The overall braking decelerations on the wet Surface Two shall not be lower than five percent less than for similar runs with the anti-locked braking system disconnected (if such a system is fitted in the car).

Partial Braking

Overall braking decelerations from 30, 60 and 90 km/h to rest shall also be measured with any one hydraulic system inoperative and an overall deceleration of 0.4 g shall be exceeded with a pedal force of not more than 500N. Any deviation shall be contained within a 3.6 m wide lane, without the need for large steering corrections. A similar performance is required in the event of a failure of servo assistance. The handbrake shall hold the stationary car from moving when facing both up and down a hill with a slope of 1 m down in 4 m along the road surface.

Brake Fade

The braking shall be free from fade to the extent that the fifth repeated braking stop, in quick succession from 110 km/h to rest, shall exceed an overall deceleration of 0.7 g with a pedal loading of less than 900N.

HANDLING REQUIREMENTS WHEN BRAKING NOT IN A STRAIGHT LINE

"Split Coefficient" Surface Test

Apply the brakes at 60 km/h with the wheels on the wet Surface One on one side and on the wet Surface Two on the other side of the car. The overall deceleration shall be better than that for the car in standard form when braked on the wet Surface Two. It shall not be difficult to control the car during the "split coefficient" test.

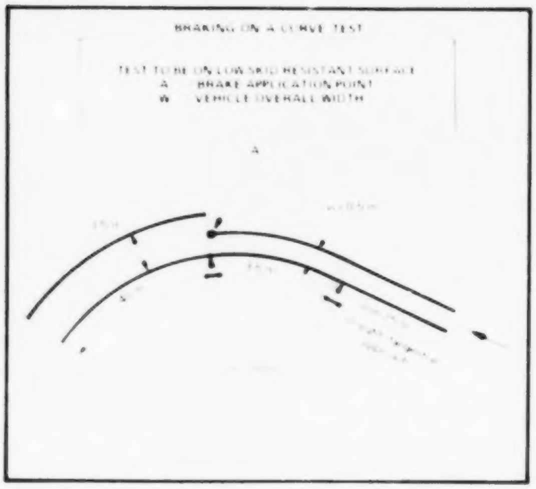
BRAKING ON A CURVE TEST
TEST TO BE ON LOW SKID RESISTANT SURFACE
A BRAKE APPLICATION POINT
W VEHICLE OVERALL WIDTH
A

Figure 1(a)

Braking On Curve

Establish the maximum steady speed at which the 50 m inside radius of the test curve shown in Figure 1(a) can be negotiated without excessive slip on the Surface Two when wet. Apply the brakes on a similar run at a steady speed 90 percent of the above. The car must stay within a 3.6 m lane while coming to rest.

Braking During Lane Change

This is to be carried out on Surface Two when wet, the configuration of the test area being as shown in Figure 1(b). Establish the maximum steady speed at which the hazard can be negotiated without braking. Find the highest speed at which two runs can be completed when braking at point A and attempting to steer through the course. The vehicle must remain within the course boundaries.

Variable Coefficient Surface

Apply the brakes at 60 km/h, 10 m before reaching the junction between the wet Surface One which is being traversed and moving on to the wet Surface Two. There must be no deviation outside a 3.6 m lane, or other undesirable effect.

FAILURES AND WARNINGS AT FACIA AHEAD OF DRIVER

It is required that any anti-locking brake system fitted shall have fail-safe characteristics incorporated in their design so that the braking reverts to the basic braking system if there is a fault in the anti-locking system. A brake warning light shall be fitted to warn the driver of one or more of the following faults.

1. Hydraulic failure or low fluid level

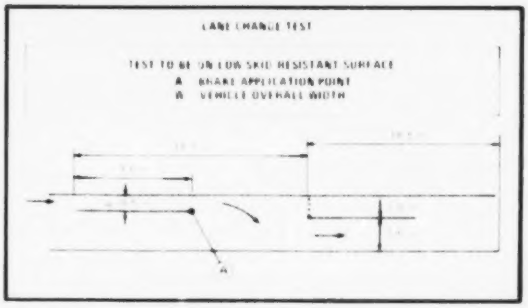
LANE CHANGE TEST
TEST TO BE ON LOW SKID RESISTANT SURFACE
A BRAKE APPLICATION POINT
B VEHICLE OVERALL WIDTH
A

Figure 1(b)

2. Failure of anti-locking system
3. Failure of servo assistance
4. Excessive pedal travel
5. Worn brake pads or linings

The levels in the fluid reservoirs shall be visible from the engine compartment.

LIFE AND SERVICEABILITY OF BRAKING SYSTEMS

The braking system shall be designed to have a life in the conditions prevailing in the United Kingdom of 50,000 miles. During this time, it should not need attention beyond possibly the replacement of friction materials and the usual servicing check to see that no unexpected damage or unusual deterioration has occurred. The brake system shall be designed so that a replacement of all critical components can easily and cheaply be carried out; it being envisaged that this would be done once during the life of a car. Automatic adjustment of the brakes to take up friction material wear shall be incorporated, in which case care shall be taken to see that the adjuster shall achieve the life set for the other braking components. Provision shall be made for direct viewing of the wear of friction materials without removing more than the wheels from the car.

TEST TRACK SURFACE FRICTIONAL PROPERTIES

The following tract surfaces are available at TRRL and form the basis of the test requirements specified for braking and handling characteristics.

1. Fine Textured Asphalt ("High Coefficient" Surface) This surface will be used dry for the braking distance tests on a high coefficient surface and wet for the better of the two surfaces in the split surface tests.
2. Smooth Mastic Asphalt ("Low Coefficient" Surface) This surface will be used wet for the braking distance tests on a low coefficient surface, the tests on a curve and the split surface tests.

HANDLING

Aims

The overall aim is to produce cars that have safer handling properties than present day cars. Since there is no proven relationship between the rate of incidence of accidents and steady-state handling and transient response behavior, no handling requirements of this type are specified. Handling requirements are to be concerned with the type of maneuver and road conditions occurring at accidents. The driver appears to cope in equally safe manner with handling ranging from that of a small car to that of a large lorry. Possibly what he is not able to cope with is sudden or unexpected changes in behavior. These may be influenced by changes in load, tire pressures, tire wear, etc.

As a consequence of this, it is felt that the following areas of car handling are important.

- Car handling should not change drastically in character when the car is used at the extremes of its range of load, tire pressure and wear, etc.
- There should be no sudden change in behavior with increase in lateral acceleration towards limiting conditions. The approach to these conditions should, if possible, be accompanied by gradual warning.
- There should be no drastic change in car lateral behavior when power is suddenly put on or removed (i.e. throttle opened or released).
- Lateral movements and changes in course, due to wind gusts, should not be excessive.
- If unbalanced forces occur at the front wheels due to differences in braking forces and adhesion, it should be possible to maintain control without the need for large steering corrections or steering effort.

TESTS AND REQUIREMENTS FOR HANDLING

The tests outlined below should be carried out under the following conditions, unless otherwise stated. Apart from meeting any particular requirements, the maximum speed at which any test can successfully be carried out, should be noted, together with the mode of failure at a slightly higher speed (e.g. drift or spin).

The car conditions shall be:

1. Normal load   driver, front passenger, two rear passengers and 50 kg load
2. Full load

   Small car: driver, front passenger, two rear passengers and 50 kg load in boot.

   Medium car: driver, front passenger, three rear passengers 75 kg load in boot.

   Larger car: driver, front passenger, three rear passengers and 100 kg load in boot.

And for both these loads, the following five tire layouts, making ten conditions for each test.

3. Run-flat tires, with not less than 75 percent tread, at design pressures
4. As for 3, but with a rear tire running flat
5. As for 3, but with a front tire running flat
6. As for 3, but with front tires worn down to legal minimum tread
7. As for 3, but with rear tires worn down to legal minimum tread.

The tests shall be carried out on Test Surface One, or on a similar surface. Outriggers may be fitted on both sides of the car to prevent overturning if it seems possible that this may occur during any of the tests.

Tests For Braking On Curve

The tests described for "Braking on a Curve" shall be extended to include the conditions specified above on the surface one when wet. The maximum speed should be noted for the runs with and without braking and the corresponding modes of failure should also be listed. For runs with conditions "4" and "5" the outer tire should be flat. No requirements are laid down, but the tests should demonstrate to what extent the aims (no drastic changes in handling behavior) have been achieved.

Tests On "Split Coefficient" Surface

The tests described previously shall be extended to include the conditions specified above on the surfaces One and Two when wet. An additional requirement is that there should be no large differences in behavior of the car, whether the steering is held rigidly or lightly. For runs with conditions "4" and "5" the tire on the higher coefficient surface shall be the flat one.

Tests Of Effect Of Sudden Power Change

The tests described for braking during a lane change shall be repeated for the conditions specified above, without braking on both wet and dry surfaces, the fastest uniform speeds for satisfactorily completing the manuever being noted. Similar runs shall then be made at the same speeds, but with the throttle being completely and very quickly released, immediately the steering has been turned for leaving the entry lane and well before the steering has been reversed to keep the car in its intended course. The test is to be repeated with a flat tire in all positions. There must be no loss of control.

Wind Gust Test

The car shall be driven through a transverse wind gust at constant speeds of 50 and 100 km/h for all the car conditions on a dry surface. A suggested gust is 35 m long with a mean wind speed of 60 km/h. The driver is allowed to correct the car's path, but he should be able to keep the car on its course with a deviation of less than 0.5 m.

LIGHTING AND VISIBILITY

Lights And Lighting

The car lighting system shall be designed on the basis of the UK *Construction and Use* requirements, though modified by anything stated in this section. The ECE and EEC specifications shall be followed for aspects not defined by the above, unless these requirements are contrary to current practice, then the matter will be settled by further discussion. The outboard angles for the side and rear lights and for the direction indicators shall be between 100° and 110°, rather than the 80° stated in existing standards.

The first flash of the direction indicators shall be clearly visible outside the car within one-fifth of a second of the driver operating the switch. The flashing frequency shall be maintained, within the specified limits, at all states of charge of the battery sufficient to start the car. The direction warning indicators shall incorporate a four-way flasher unit for emergency use.

A two-level brake light system shall be installed, which consists of two brake lights on each side of the rear of the car. The outer pair shall meet all normal requirements. The inner pair shall be adjacent to and directly inboard of the outer pair. The inner pair shall light up when the braking deceleration reaches about 0.4 g, being operated by a pressure transducer in the brake line or by other means. The two pairs shall be of the same brightness and of even illumination over their rear faces, which shall each be at least 4,000 mm^2 in area. The outer brake lights shall lock-on when the brakes bring the car to rest and shall be switched off either when the car moves off again or when the ignition is switched off.

The headlights shall be fitted with headlight wipers and automatic dimming in the presence of oncoming lights. If the change in attitude of the car between driver alone and full load is such that the headlight aim is deemed to be unacceptable on discussion between the contractors and TRRL, it shall be fitted with a means of headlight levelling.

The equipment shall include a reversing light containing a bulb or bulbs of a total wattage of not more than 36 watts. The driver, while in his normal seating position, shall be made aware of failures of the side, rear and outer brake lights by one of the following means:

- A fiber optics system providing a direction indication at the fascia
- A failure indicator unit with a single warning light at the fascia
- Telltales which can be viewed directly from the edge of the side light, or viewed through the external mirrors for the rear and brake lights.

Visibility

The view available to the driver is dependent on the positions of the obstructions to his view and also on the cleanliness of the glass and mirrors used by the driver. The following requirements supplement, or, if they differ, replace, the basic requirements stated in UK *Construction and Use* and EEC standards and expected future EEC and ECE legislation.

The restrictions to view presented by the A posts shall be within that proposed in the ECE draft and shall be no greater than those of the cars on which the demonstration safety cars are based. Provision of a good field of view shall be made for unusually short or tall drivers by means of seat adjustment, which may include the use of special spacers replacing the normal means of seat adjustment.

Consideration shall be given to the means of reducing sun dazzle when the sun is low in the sky, by choice of the angle of rake of the windscreen and by means of efficiently washing and wiping it.

Means shall be provided for washing and wiping the most critical area of the rear window for the rear view when using the interior mirror. Rear view mirrors shall be provided in accordance with the EEC Regulation 71/127/EEC. Two external mirrors, as well as an internal mirror, shall be provided to give a good view to the rear. The external mirrors shall be viewed through a part of the windscreen which is covered by the normal windscreen wipers. The defrosting and demisting requirements for the windscreen are those drafted by ECE and ISO.

Consideration shall also be given to the means of demisting the interior faces of other glass around the passenger compartment when it is being driven in a temperature of 0°C after being parked when dry and left long enough to cool down to that temperature and when there is only a front seat passenger and driver. It is suggested that a sufficient flow of air through the passenger compartment be provided under the control of the driver, combined with auxiliary means of warming the side and rear glass by means of electrical power or ducted air, so that the most critical 20 percent of the areas of glass for visibility at intersections and to the rear, be clear within five minutes of entering and starting the car.

AIDS TO DRIVING

The following comments are in the form of a working document rather than a specification.

- A head-up display for the speedometer reading is required.
- Provisions should be made for the installation of the following items of equipment. Details are available from TRRI.
- Space should be left for the radio part of a RITA installation for the audible receipt of messages from a control center. This requires a standard radio opening and a depth of 200 mm beneath it should be left free.
- A log throttle, which leads to smooth throttle control at low speeds may be fitted.
- Cruise control of speed and the car borne components for compulsory speed limit control may be required to be fitted for demonstration purposes.
- Automatic following of vehicles ahead and braking control may also be included. This is based on data, which derives from a radar set in the front of the car; it also needs space for an electronic unit and actuators within the engine compartment.

The positioning of all controls and warning lights shall be considered in relation to the ergonomics of the driving task, with particular reference to the need for the driver to reach, when restrained by the belt system, the essential driving controls, the heater and ventilation adjusters and preferably the nearside front door latch.

Some warning device indicating that a tire is flat shall be provided.

DESIGN FOR DURABILITY AND LOW OPERATIONAL COSTS

It is intended that the design and quality of production of the cars called for in this specification should be up to the best of cars currently being produced in large numbers, with particular attention being given to the following features.

The cars shall be designed for safe operation for 100,000 miles or ten years, whichever is less. Protection against rusting of the body and other corrosion shall be provided so that cars would be expected to pass the present annual vehicle inspection tests during this period without the need for structural repairs. Brake durability requirements are also included.

The tires selected shall be of such a size that they are not overloaded at either the front or rear axles when the cars are fully laden.

The vehicle suspension shall be such that there is no undue tire wear and the tires shall be selected for long wear within the limitations imposed by the handling requirements.

Despite the design restrictions placed by the occupant protection and other safety requirements, attention shall be given to the need for ease of maintenance and the replacement of faulty units.

The design shall be such that the cost of repair of minor accident damage is low, taking into account both the costs of replacement components and labor charges. Minor impacts shall not lead to damage extending beyond the front and rear bumpers into the body work or lighting assemblies. Some form of side bumper strip should be fitted at bumper height to absorb side scratch type contacts from cars with a matching bumper layout.

APPENDIX II

CAR FRONT END DESIGN AND PEDESTRIAN SAFETY

The trajectories of anthropomorphic dummies struck by passenger cars have been studied with a view to determining a front end vehicle configuration which will alleviate pedestrian trauma. The study has shown that the leading edge of the bonnet must be below the height of the center of gravity of the pedestrian dummy in order to prevent it being suddenly accelerated and projected on to the road ahead of the vehicle.

Provided that the leading edge of the bonnet is sufficiently low, impact of the vehicle against the legs of the pedestrian causes the torso to fold over and the head to strike the bonnet or the windscreen. For a brief period of time vehicle and pedestrian then travel with the same forward velocity. In order to pick up a child dummy the leading edge of the bonnet must be considerably lower than the height which is typical of current large volume production cars. Bumper height should also be lower than the current practice in order that the first contact with the legs of an adult pedestrian is below rather than at the knee.

The severity with which the head of the pedestrian strikes the bonnet can be reduced by markedly rounding the front end of the vehicle so that in most impacts the pedestrian is projected at an angle across the bonnet. Overall bonnet length should be sufficient to minimize the risk of the head striking the windscreen. Figure 1 shows an experimental front end embodying the features described. On current vehicles the junction of the scuttle and the base of the windscreen has considerable stiffness. It could be made more flexible by continuing the screen well

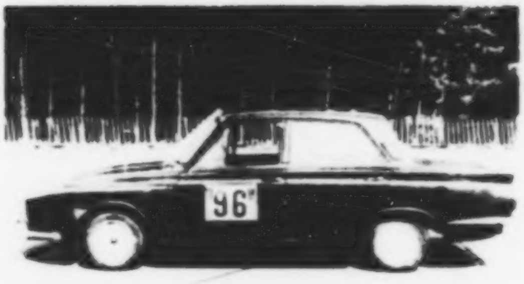

Figure 1

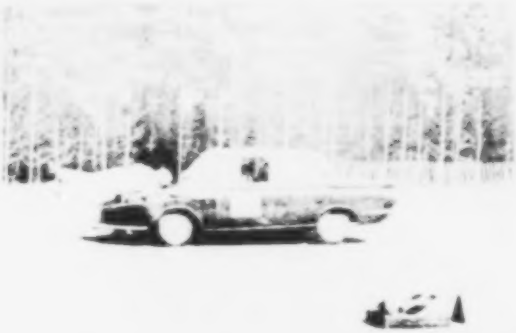

Figure 2

below the scuttle leaving a gap between the two. This treatment also has the advantage that the pedestrian is prevented from striking the lower edge of the screen thereby reducing facial injuries. The foregoing features would tend to reduce the severity of pedestrian injuries sustained in the initial stage of the accident during which the pedestrian is projected on to the vehicle. However, the pedestrian rarely remains upon the vehicle but falls off either in front of or to one side to strike the road with a severity which increases with the speed of the vehicle. The worst situation arises when the vehicle brakes are not applied until after the impact occurs, the pedestrian is then projected forward of the vehicle with the velocity attained while lying on the bonnet and strikes the road with considerable force. Some means whereby the pedestrian is retained on the vehicle until it is brought to rest is therefore desirable.

One such device consists of a metal tube shaped to conform to the periphery of the vehicle (Figure 1). In its rest position in the tube itself forms the leading edge of the bonnet, in practice it would be covered with an elastomer material to reduce pedestrian injury. Impact with a pedestrian raises the tube to a height sufficient to prevent the pedestrian from sliding off the vehicle (Figure 2). The pedestrian catcher device is the subject of a provisional patent specification.

THE UNITED KINGDOM TECHNICAL PRESENTATION

SECTION **TWO** PART **4**

THE U.K. APPROACH TO ANTHROPOMORPHIC DUMMY SPECIFICATIONS

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INTRODUCTION

When looking objectively at all ESV programmes, the underlying purpose of the work is to save human lives. Therefore, any safety vehicle or system developed and deemed successful must save the life or prevent serious injury to the person involved.

However, the thought of using human guinea pigs to test systems until they have been proven thoroughly, is not one to be relished. It is, therefore, obvious that satisfactory models of humans must be made, to behave as much like humans as is reasonably possible, to enable the systems to be developed to suit the real world environment.

In Britain, it is felt that these simulations or dummies should react, as far as possible, in a manner compatible with results found by research into "real world" accidents. For instance, it has been found, repeatedly, that people wearing 3-point seat belts in accidents do not sustain the serious injuries predicted by the results of accelerometer readings from test dummies. A basic dummy design philosophy has, therefore, evolved which attempts to relate dummy design and performance to real life situations.

A BRIEF LOOK INTO THE DUMMY INDUSTRY IN THE U.K.

Anthropomorphic dummies have been made in Britain since 1954, when the Royal Aircraft Establishment at Farnborough developed a robust and simple dummy for parachute testing. This dummy has undergone several developments and is now a more sophisticated version of the same simple dummy. It now has a means of adjusting limb frictions, but no reproduction of chest response, other limb areas, the pelvis for instance, are still non-representative and unsuitable for compliance testing of automotive restraint devices.

Research and development was started in Britain for the construction of a more lifelike dummy in 1968 by the Motor Industry Research Association. In particular, the response of the chest and the shaping, and operation of the pelvis were studied, as both parts played an important role in assessing belted restraint systems.

In 1970, David Ogle Ltd., and MIRA agreed to form a partnership to further develop and manufacture the dummy originally presented. This was a logical expansion of Ogle's interests as car designers into the advancement of car safety. The first Ogle dummy, the M50/71, was designed to comply with the American SAE J963 specification for dummies, together with the newly introduced FMVSS 208 pelvis shape requirements for 50th percentile male dummies.

At that time, the amount of data was limited and compromises had to be made to establish the exact dimensions and weight and shapes of the various components. As a result, a dummy was designed to conform to the J963 specifications in terms of dimensions and weights and ranges of motion.

The major departures from conventional practice were:

1. The design response of the chest was such that results of chest impacts correlated to cadaver information much more closely than other dummies (Figure 1). This response was originated by carefully matching the spring rates of the ribs with the effects of masses representing chest contents and shoulder musculature. This design work was pioneered by John Searle and Christine Haslegrave of MIRA and production chests have been in used worldwide during the last 18 months. These have shown that the response of the chest is the most humanlike development to date.
2. In connection with the chest simulation, the sterno-clavicular joint and the joint at the acromium process were carefully developed so that a clavicle of near human form articulates in the correct manner and allows a more accurate assessment of the restraining properties of 3-point diagonal belts.

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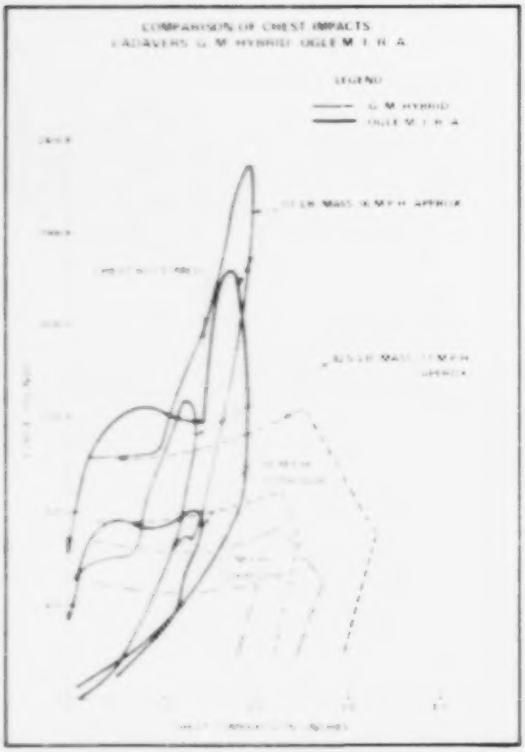

Figure 1

3. The other major area of departure from convention was in the pelvis joint and the shaping of the femur. At the pelvis neck joint, much effort was put into achieving a full range of motion found in life and a special internally-retained ball joint was developed which allowed this full range of motion.

Special attention was paid to the shaping of the femur. S8.1.15 of FMVSS 208 states "A load sensing device is installed in each upper leg, 4.25 inches from the knee's axis of rotation so that all force transmitted from the knee to the upper leg is measured." The form was unspecified in both 208 and J963, in consequence, the shaping of this component was left to the individual manufacturer.

A more representative shape was evolved which had a proper femur-neck and an angled knee. A simple check on the effectiveness of this feature is to attempt to cross the legs of a conventional dummy and compare them with the legs of the M50/71 and yourselves. You will see that if the leg of the conventional dummy is straightened, the angle it makes is totally unrepresentative.

THE NEED FOR NEW DUMMY SPECIFICATIONS

As part of the U.K. ESV programme, a research contract has been negotiated between the U.K. Government and a motor manufacturer to produce a range of vehicle prototypes. These vehicles will incorporate as many safety features as is possible, the principal candidates being those found to be practical and economically justifiable in terms of injury prevention. The trade-offs involved here are the probable additional cost features on each car.

As a basis for this contract, a series of performance requirements have been drawn up, both for primary safety or accident-avoidance features and for secondary-safety or injury-reducing features. For the latter requirement, it soon became apparent that it would be necessary to specify closely the dummy to be used in the various impact tests called up and also that no current dummy was suitable for such an application. At this time, there were only two specifications for anthropomorphic dummies in existence, the SAE J963 and the American DOT purchase specification for compliance test dummies. So that there would be a useful exchange of data, the U. K. Government decided that it would take these others as a base for its own specification and expand upon them where necessary.

DESIGN CRITERION

The criterion on which these additions and modifications to the existing specification were based incorporated the main points of the DOT purchase specification but with certain additions and modifications in terms of form, performance, and construction.

Our modified specification was based upon the surmise that dummies should, as far as possible, be:

- Humanlike in their behavior.
- Able to produce repeatable results under similar test conditions.
- Robust and suitable for repeat testing.
- Economical in use.
- Built to comply with a performance specification.
- Easy to adjust and set up for testing.
- Calibrated against known real life accident data.

As a desirable addition to these criteria, it was requested that the dummy should be able to stand upright for pedestrian work.

After considering these implications, it was decided that it would be necessary to produce a design standard for the dummy, in addition to the performance standard. This is in accord with current international standards practice where, although performance standards are generally favored for safety features, the actual design of test devices used to assess the performance is usually specified. The outline specification for the "Occupant Protection

Assessment Test Dummy" is given in Appendix 1, but it is intended that this will be supplemented with a complete design specification and various detailed performance requirements resulting from the current development programme.

WORK PROGRAM

Work has been in hand since September 1972 at Ogle, who was chosen as the contractor on the project because of its experience and capacity in the design and manufacture of anthropomorphic test devices. The Transport and Road Research Laboratory has been sponsoring the work and controlling the project while MIRA has been carrying out test programmes and advising on design aspects. We believe that the combination of skills from a Government Research Laboratory, the Motor Industry and a dummy manufacturer should combine to make a practical and serviceable dummy.

GENERAL

The external form of the dummy will be based on the American SAE "Golden Shells" when this data becomes available. The intention is to simplify the geometry of dummy to yield results which are less sensitive to minor variations in orientation on impact. For example:

The Head The head form is not yet finally decided but is likely to be of a form such as that shown in Figure 2. The flexible covering will be a uniform thickness over a stiff skull which, together with a more uniform shaping of the head, will lead to more repeatable results where the head makes contact with objects inside the car. The internal form of the skull is designed to make the structure stiff and light, so as to give more repeatable results.

The Neck The neck is of an entirely conventional rubber form, with a special mix of rubber which, to date, has been found to give the most lifelike and repeatable results of the alternatives that have been tried. There is much research being carried out in Britain into neck performance, and as new data becomes available it will be incorporated into the design.

The Shoulders The shoulders and clavicle are a more robust and simplified form of the Ogle M50/71 dummy. It can be seen from Figure 3 that the metal-to-metal sliding has been eliminated and that a more definite control of motion is achieved by the use of representation of the trapezius muscle area. Motion of the component is controlled by a rubber bump stop. The shoulder joint is of similar construction to all those throughout the arm and consists of a split, threaded-plastic bush which retains the turned steel pivot. Friction is controlled by tightening a pinch bolt. This bolt also controls the range of motion of the joint (Figure 4). It can be seen that this type of joint eliminates metal-to-metal contact (a major source of spurious acceleration signals), is easy to adjust and eliminates end and side float at the joint.

The Chest The chest unit is an updated version of the M50/71 unit incorporating all the experience of the past 18 months. The basic design features are:

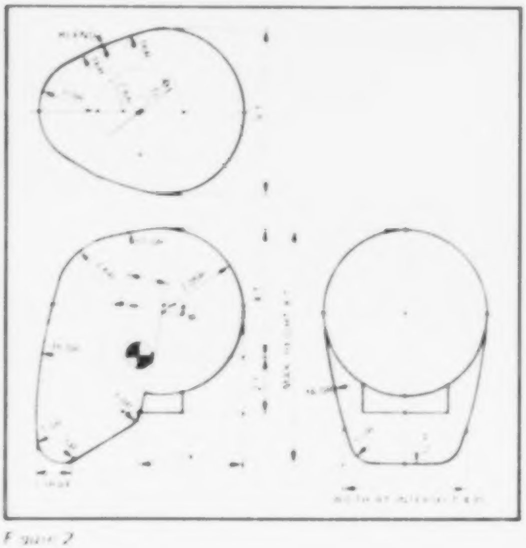

Figure 2

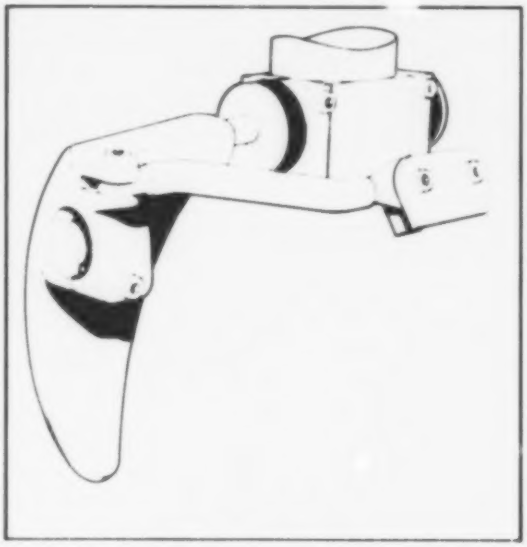

Figure 3

Figure 4

1. The form of the human rib cage is followed as closely as possible so that when the cage is struck by external force, the locus of movement is as close to human as possible. The upsweep from front to back on each rib form is 10°.
2. The static stiffness of the ribs is matched very closely to that found in cadavers. The ribs are located at the spine so that the rib cage as a whole hinges backwards.
3. To enable the effective stiffness of the chest to increase during impact, simulations of the lungs and internal organs are allowed to float freely in close proximity to the wall of the chest. Additional dynamic stiffening comes from the effects of the mass of the shoulder blades moving forward under impact and supporting the upper rear part of the rib cage (Figure 5).
4. Experimental chests have been made using plastics for the rib cage to eliminate some of the "ringing" effects found to some extent when using steel ribs.
5. Investigations are also being carried out with regard to the possible use of materials other than the current lead and polyether foam pack used to simulate the internal organs.

The Lumbar Spine The human lumbar spine is an area which can be compared superficially at least with the neck structure in terms of construction and action. Experimental dummy spines, which are adaptations of the neck structure but with stiffer rubber compounds, have been made. Slightly more sophisticated versions have been fabricated which have a central steel tie connected at each end to ball joints.

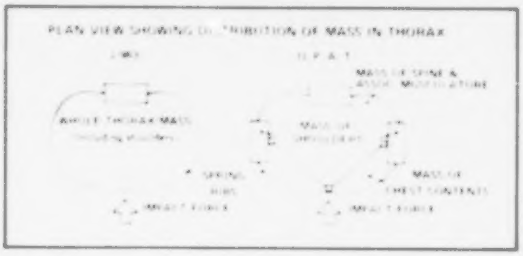

Figure 5

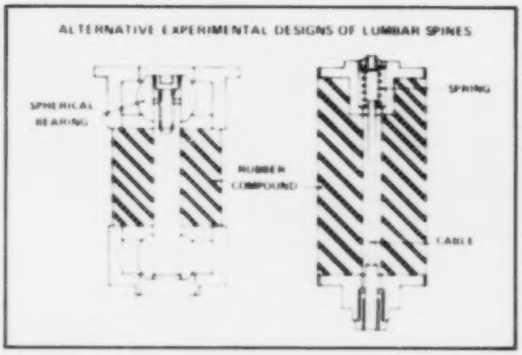

Figure 6

Both of these alternatives have the major advantage of not requiring any setting and have been designed to the static performance requirements of the American purchase specification (Figure 6).

The Pelvis The form of the pelvis conforms with FMVSS 208 and has been adapted to suit a more robust overcenterball, hip joint representation, which differs in construction from the undercenterball found in humans. The friction is now controlled by a pinch bolt operating on the external diameter of a ground ball joint. The femus is attached via a "T" head bolt system, which allows the ball to be precision ground and a close tolerance maintained on the fit of the femur to the ball.

The Femur/Knee The femur is shaped in a lifelike manner as specified below. The ballast weight which has to be carried to obtain the correct weight for this limb has been specially shaped so that if the femur strikes and slides under a dashboard or along the underside of a steering wheel, the "bone" structure does not offer unnaturally sharp corners which may affect the movement of the dummy.

Provision has been allowed for the mounting of a standard load cell 4.25 inches back from the pivot of the knee joint. (to comply with FMVSS 208 Figure 7)

The lower leg incorporates at its upper end a steel hemisphere of 2¼-inch diameter which represents the knee cap. This was thought to be an important innovation, inasmuch as any minor variations in the angle at which a rectilinear type knee form strikes a dash, or any other object within a car, can affect the repeatibility of the reading at this joint (Figure 8).

A vinyl-covered foam flesh system, with the exception of the chest and knee areas, will be permanently attached to the structure of thedummy, with

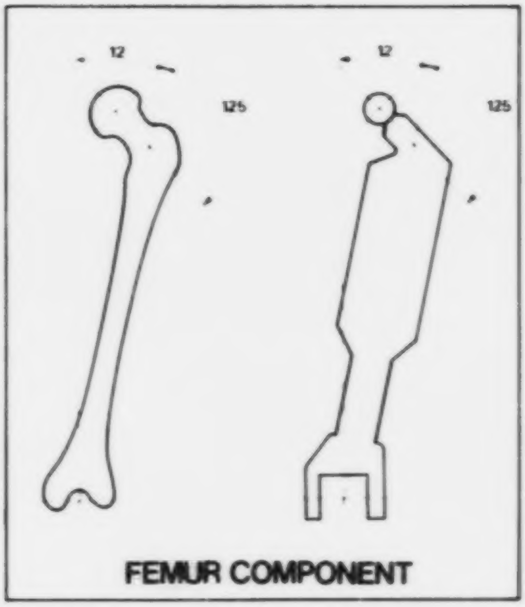

Figure 7

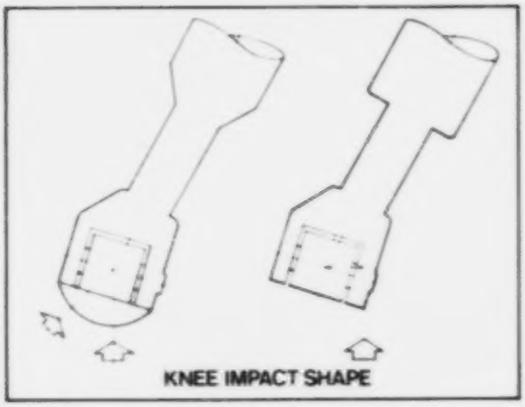

Figure 8

provision made for internal wiring. The abdominal sack of the dummy is made of a flexible solid plastic material, but consideration is being given to the use of fluid filled sacs for pressure measurement.

SUMMARY

In the United Kingdom, the ESV programme has shown a need for a fully specified, repeatable, robust and humanlike compliance test dummy. At the same time, while producing a dummy to its own specification, the Transport and Road Research Laboratory is keeping abreast of other dummy research work, so that the final specification will be suitable for testing to existing and future occupant protection standards.

The development work to date has proved that it is possible to attain practical standards of humanlike performance, combined with relative simplicity, robustness and economy of manufacture. In particular, the concept of producing a more simple and robust dummy, which is less sensitive to minor variation in setting up procedures and test parameters, will produce benefits in terms of repeatibility of test results. The final specification will be drawn up as soon as the full calibration programme has been completed.

DRAFT OUTLINE OF PROPOSED SPECIFICATION FOR OCCUPANT PROTECTION ASSESSMENT TEST DUMMY

PURPOSE AND SCOPE

This document describes the 50th percentile anthropomorphic test dummy which will be used for compliance tests for cars manufactured to proposed motor vehicle safety requirements. This test device is designed to be used in the evaluation of protection systems for vehicle occupants during real and simulated impact conditions.

TERMINOLOGY

1. The term "dummy" refers to the test device described by this section.
2. Terms such as "head," that normally refer to parts or areas of the human body, refer to the portions of the dummy that correspond to those parts or areas as they are commonly defined.

GENERAL DESCRIPTION

1. The dummy configuration corresponds to that described in Appendix 1 (not contained in this publication). General dimensional data is reflected in Table 1.
2. The dummy also conforms with the following additional specifications:
   a. The assembled dummy distal segments are capable of motion about effective centers of rotation relative to adjoining segments.
   b. Each structural segment of the dummy, except for the hands, feet and spinal column, when subjected to a tensile force of 400 pounds does not elongate more than .5% of the segment length and returns to its specified length when the force is removed.
   c. The spinal column shall represent the compressibility and extensibility of the human spinal column by complying with requirements to be specified at a later date.

TABLE 1

General Dimensional Anthropomorphic Data for 50th Percentile Adult Male O.P.A.T. Dummy

Centers of Gravity (± 2½%)	**Values in inches**
Head (forward from backline of body)	5.6
Head (below top of head)	4.7
Shoulders (forward of backline)	3.8
Shoulders (below top of head)	14.1
Abdomen (forward of backline)	4.9
Abdomen (below top of head)	20.8
Buttocks (forward of backline)	5.3
Buttocks (below top of head)	31.2
Head and trunk whole (forward of backline)	4.7
Head and trunk whole (below top of head)	22.7
Circumferences (± 2½%)	
Head	* 22.5
Neck	15.0
Chest	* 37.7
Waist (sitting)	* 33.0
Buttock (sitting)	41.5
Thigh	22.4
Lower Thigh	17.3
Knee (sitting)	15.4
Calf	14.4
Ankle	8.9
Arm (at underarm)	12.4
Elbow (flexed)	12.2
Wrist	6.8

Segment Weights (± 2%)	**Lbs.**	
Head and neck	11.2	
Upper arm - each	5.4	
Forearm - each	3.4	
Hand - each	1.4	
Upper leg - each	17.6	
Lower Leg - each	6.9	
Foot - each	2.8	
Shoulders and upper thorax	17.3)	
Lower thorax and upper abdomen	23.0)	see
Lower abdomen, buttocks and	)	below
upper thighs	37.5)	
TOTAL DEVICE WEIGHT	**164**	

The weight of the torso is distributed as follows:

Shoulders and thorax (i.e. rib cage)	37 lbs.	made up of
Weight of movable part of shoulder (ea.)	5.4 lbs.	
Weight of thorax	19.2 lbs.	
with the additional weights of (each)	3.4 lbs.	placed

on the chest wall adjacent to each movable shoulder in order to simulate the muscle. Of the thorax weight 6.3 lbs. simulates the movable chest contents with a forward movement of up to 2 inches relative to the chest wall.

* requirement of SAE J963

d. Adjacent segments are joined in such a manner that throughout the range of motion and also under crash impact conditions, there is no impact contact between skeletal elements, except for those contacts which are already in existence under static conditions.

e. Structural properties of the dummy are such that the dummy conforms to every provision of the specifications both before and immediately after the test for which it is used.

f. The external shape of the flesh of the dummy shall be based on the SAE Golden Shell.

COMPONENTS

1. *Head*

a. The shape of the head shall be based on a geometric simplification of the 50th percentile "Golden Shell" developed by SAE and shall exclude nose and ear representation. The head is composed of a rigid skull which is covered with a resilient skin whose normal thickness is 3/8 ± 1/16 inch, and whose mirror-image sectional depth variations about the midsagittal plane do not exceed 1/16 inch. The skin is mechanically attached to the skull to prevent shifting or separation during impact.

b. The skull cavity has an internal horizontal bulkhead for mounting an accelerometer cluster whose centroid of seismic masses is located 7.5 ± 0.1 inches above the neck-to-torso attachment surface and 3.9 ± 0.1 inches forward of the back of the head. The bulkhead may also be used for the attachment of the head to the neck.

c. A photometric target at least one inch in diameter is affixed to each side of the head's outer surface. The target is centered on the transverse axis going through the CG of the head in a manner to reflect the true motion of the skull during the impact. A vertical line following the contours of the head at least six inches long and coinciding with the CG of the head is defined on both sides of the head's outer surface by photometric targets.

d. When tested in accordance with the procedure of this paragraph, (items (1.) through (4.)), the head response curve (acceleration vs. time) is unimodal in shape, attains peak accelerations of less than 250 G's in ten inch drop heights and less than 790 G's in 35 inch drop heights, and whose pulse duration at 100 G's level is between 1.0 ms and 1.5 ms. These figures will be more closely defined in the light of test results.
   1) The head is suspended in the configuration shown in Figure 1.
   2) The impact point is located on the head at the intersection of midsagittal plane with horizontal plane located two inches below the top of the head.
   3) Free fall drops are made on a rigidly supported flat-steel plate at least two inches thick, and two feet square, from designated heights measured from intended points of impact on the head to the block.
   4) The head resultant accelerations are measured by instrumentation specified in the instrumentation section of this document and compared with the values specified in this requirement.

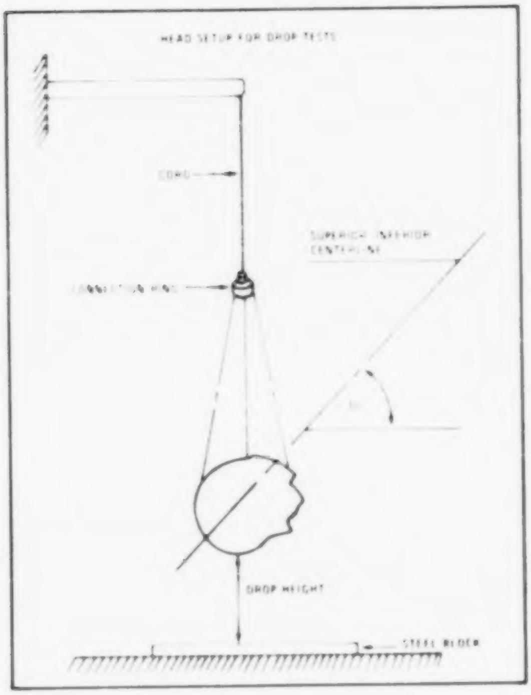

Figure 1

2. *Neck*

a. The neck is rigidly mounted to the horizontal bulkhead of the skull (see Section 1b) and to the mounting plate on the torso whose center-point is located 20 inches above the hip pivot center as measured on an upright seated dummy.

b. The mounting of the neck at the torso is capable of angular adjustment to allow alignment of the head relative to the vertical for various seat back inclinations.

c. Under applied static loads, the head-neck complex, as attached to the torso in an upright seated position, is capable of motion within the ranges specified in SAE J963. Upon removal of load, the head returns to the original set-up position.

d. The neck, when subjected to dynamic tests specified in this paragraph (items (1) through (4)) has the following dynamic response as measured at the CG of the head, while the head resultant acceleration does not exceed 30 G's.

	Head Rotation Relative to Initial Position (Degrees)	Time From Point of Impact (ms ± 4%)	Chordal Displacement From Initial Position (Inches ± 0.5)
Forward loading use	0	0	0
	30	27	3.1
	60	41	5.7
	69 MAX	55	6.8 MAX
Forward loading use	60	71	5.7
	30	88	3.1
	0	101	0

The head-neck complex may hyperextend during the rebound phase without exceeding maximum angular rotation and chordal displacement of 60 degrees and five inches, respectively.

1) The head-neck complex is mounted on a rigid pendulum as shown in Figure 2, the head-neck midsagittal plane coinciding with the pendulum vertical centerline and pendulum motion plane, and, the head-neck superior-inferior axis remaining parallel but offset by .7 inches, from the pendulum center line.

2) The free-falling pendulum is decelerated from tangential velocity of 26 feet per second, as measured 65.2 inches from the point of pendulum pivot, with a pulse having an amplitude of 20-26 G's for a total time duration of 40/45 ms and rise and decay time each not exceeding 5 ms. The head-neck complex is allowed to flex freely without contacting an external surface.

3) The acceleration time history of the pendulum is measured with instrumentation having frequency response of the channel class 60, SAE J211, Recommended Practice, dated October 1970.

4) Measurement of the head acceleration conforms with the instrumentation requirements described in this document. The angular and linear deflection time history of the head CG is determined either by high-speed photography at a film speed not less than 1,000 fps, or by other appropriate instrumentation, which permit head motion analysis in increments of 1 ms.

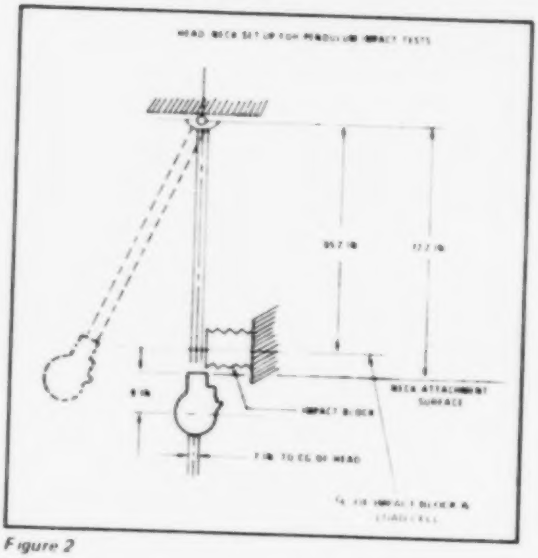

Figure 2

The neck requirements will be aligned more closely with the Mertz and other data in the future.

3. *Thorax*

a. The thorax is composed of the rib cage and the shoulder assemblies which have provisions for the attachment of the arms, the neck and the lumbar spine.

b. It has provisions to rigidly attach the neck at a point 20 ± 0.1 inches above the hip pivot, as measured at the centerpoint of the bottom surface of the neck of the dummy in an upright seated position.

c. It has provisions to attach accelerometers to the rigid spine so that the centroid of the tri-axial instrument cluster is in the midsagittal plane 5.7 ± 0.1 inches below the neck to the thorax attachment surface centerpoint.

d. The sternum of the thorax is capable of at least three inches movement, relative to the posterior part of the rib cage in the anterior-posterior direction, without causing contact between dummy parts that are not in contact in the undeflected position.

e. When tested in accordance with the procedure outlined in this subparagraph (items (1) through (4)) the thorax has the following static performance:

Vertical Test Point Location	Transverse Test Point Location	Load Deflection Rate 1 bs./inch 0-1000 lbs. 0 - 3
15 inches below top of head	Midsagittal plane	<600
18 inches below top of head	Three inches either side of midsagittal plane	< 600

1) The anterior portion of the rib cage test area is within rectangular bounds extending six inches on each side of the midsagittal plane and between 12 and 21 inches below the top of the head.

2) A rigid test probe, having a flat circular face 6 ± .01 inches in diameter with the edge reduced to .5 ± .01 inches is used in load deflection tests.

3) With the dummy supported at its back against a flat and a rigid test surface, the test probe is pressed in the anterior-posterior direction into the segment being tested at a rate not exceeding one inch per second, the

face of the probe being parallel to the test surface at the instant of contact and its logitudinal centerline coinciding with the designated test point.

4) Probe force vs. deflection relative to the spine is recorded and compared with specified values.

f. The thorax, when tested in accordance with the procedure outlined in this subparagraph (items (1) through (5)), has the following dynamic response:

Impact Velocity (Feet per Sec)	Maximum Force (lbs)	Thorax Defl. (Inches)	Min. Internal Hysteresis (%)
14	900	1.5-2.0	50
22	1600	2.0-3.0	50

1) The complete dummy is seated on a hard, horizontal surface, with its head, neck and torso in an upright posture, unsupported and unrestrained and the limbs extended horizontally as shown in Figure 3.

2) A rigid test probe, having face dimensions specified for the static test in item e(2) and weighing 50 pounds, including instrumentation, impacts the dummy's chest at the designated speed in the anterior-posterior direction. During the impact, the probe is restrained, from any other motion but translation, in the anterior-posterior direction.

3) The chest is impacted in the anterior-posterior direction at the point in midsagittal plane which is 18 inches below the top of the head, the probe face being parallel to the impacted surface and its center coinciding with the specified impact point.

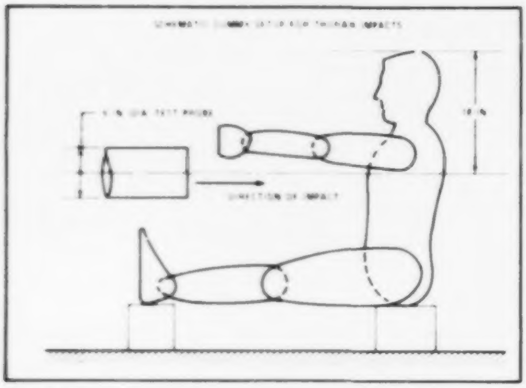

Figure 3

4) The time history of the impact force, as measured by the load cell in the impactor, versus deflection of the impact point relative to the dummy's back (spine) is recorded and force-deflection curve of the entire event plotted.

5) Internal hysteresis is measured by the ratio of the area bounded between loading and unloading portions of the force-deflection curve and the total area enclosed under the loading curve.

4. *Lumbar Spine-Pelvis*

a. The lumbar spine-pelvic assembly consists of the lumbar spine, which provides for angular motion capability between the thorax and the pelvis, of the pelvic assembly and of a simulation of the abdominal viscera.

b. The pelvic structure is of a rigid construction having provisions for upper leg attachment at the hips and for the upper torso at the lumbar spine.

c. The lumbar spine shall be designed to provide controlled motion of the adjoining body segments so that with the pelvic part of the torso and the femurs of the upright seated dummy rigidly held to a hard seat and the rest of the torso unsupported.

1) There is no more than two degrees flexion or extension of the long axis of the torso when the complete dummy is tipped 15 degrees forward or rearward from an upright seated position, and

2) There is no more than 10° flexion or extension of the long axis of the torso when the complete dummy is tipped 90° forward or rearward from an upright seated position. While in this 90° forward or rearward position, a vertical load of 90 pounds is applied at the CG of the thorax or its equivalent on the exterior surface, the thorax's angular deflection relative to pelvis increases to a minimum of 30 degrees.

d. When tested in accordance with the procedure of this paragraph (item (1) through (3)), the abdomen produces a resistance force of not more than 160 pounds at .50 inches of deflection with each .1 inch increment over 0.50 inches resulting in load increase of not more than 35 pounds, up to a total of 1½ inches.

1) The dummy is placed in supine position on a flat and rigid surface.

2) An 18-inch long, 6-inch diameter rigid cylinder is applied to the abdomen, 20.5 inches inferior to the top of the head, with the longitudinal centerline of the cylinder horizontal and perpendicular to the spine.

3) The load is applied to the cylinder in the anterior-posterior direction, while maintaining its centerline horizontal, and corresponding resistance force and probe penetration of the abdomen are recorded until specified deflections are reached. Loading rate does not exceed 0.1 inches per second.

5. *Limbs*

a. The arm assemblies consist of the upper arms, the lower arms and the hands, and the leg assemblies are made of the upper legs, the lower legs and the feet.

b. Except for the hands, each limb must have rigid simulations of skeletal elements which are interconnected with joints and each member must be covered with resilient materials simulating the flesh and the skin.

c. The knee joint must be so constructed that only a 2¼ inch diameter hemispherical kneecap attached to the lower part of the leg can contact a flat, rigid, vertical wall while the upright seated test dummy with its toes against the wall is moved in the forward, horizontal direction parallel to the midsagittal plane.

d. The femurs have provisions to mount load sensors axially whose center is 4.25 ± .02 inches from the knee center of rotation. The femur weights shall blend into the femurs and the upper surface of the femur shall approximate to a 1½ inch diameter cylinder. The plan axis of the femur in a seated position shall follow the dimensions reflected in Table 2 and illustrated in Figure 4.

TABLE 2 refers to Figure 4

Anthropomorphic Data for 50th Percentile Adult Male O.P. A.T. Dummy

Dimensions HORIZONTAL

Letter	Title		Value in inches (± 2½%)
HA	Head breadth	*	6.1
HB	Head depth	*	7.7
HC	Occiput to Z axis		1.7 (Ref)
HD	Lower neck attachment center point to Z axis		3.4 (Ref)
HE	Shoulder breadth (bideltoid diameter)	*	17.9
HF	Shoulder pivot to back line		4.1
HG	Biacromial width		15.8
HH	Chest breadth		12.0
HI	Chest depth	*	9.0
HJ	Elbow elbow breadth		16.5
HK	Waist breadth		10.6
HL	Waist depth		7.9
HM	Lumbar pivot to back line		3.3 (a)
HN	Buttock popliteal length	*	19.5
HO	Buttock-knee length	*	23.3
HP	Seat breadth		14.0
HQ	Foot breadth (maximum)	*	3.8
HR	Foot length	*	10.5
HS	Buttock section	*	10.0
HT	Elbow fingertip length	*	18.7
HU	Fingertip wrist length		7.5
HV	Elbow pivot to wrist pivot		9.9
HW	Hip to knee pivot		15.5
HX	Hip pivot to back line		5.5

VERTICAL

Letter	Title		Value in inches (± 2½%)
	Stature (standing)		68.3
VA	Sitting height (erect)	*	35.7
	Sitting height (normal)		34.1
VB	Shoulder height (sitting)		23.6
VC	Shoulder elbow length	*	14.1
VD	Shoulder pivot to seat line		21.5
VE	Shoulder pivot to elbow pivot		10.5
VF	Elbow rest height	*	9.5
VG	Hand breadth at thumb		4.1
VH	Lumbar pivot to seat line		9.0 (a)
VI	Knee height (sitting)	*	21.4
VJ	Popliteal height	*	17.3
VK	Knee pivot to ankle pivot		16.2
VL	Thigh clearance	*	5.7
VM	Ankle height (i.e. height of medial malleoulus or joint height)		3.5
VN	Hip pivot to seat line		3.5
VO	Hip pivot to centre point of the attaching surface of neck to thorax		20.0 (Ref)
VP	Top of skull to lower neck attachment center point		12.2 (Ref)
VQ	Top of head suprasternal notch		12.8
VR	Sternal length		8.2
VS	Neck length		4.5
VT	Head section	*	9.3
VU	Shoulder section	*	16.9
VW	Abdomen section	*	25.1

* requirement of SAE J963

(a) effective center of rotation

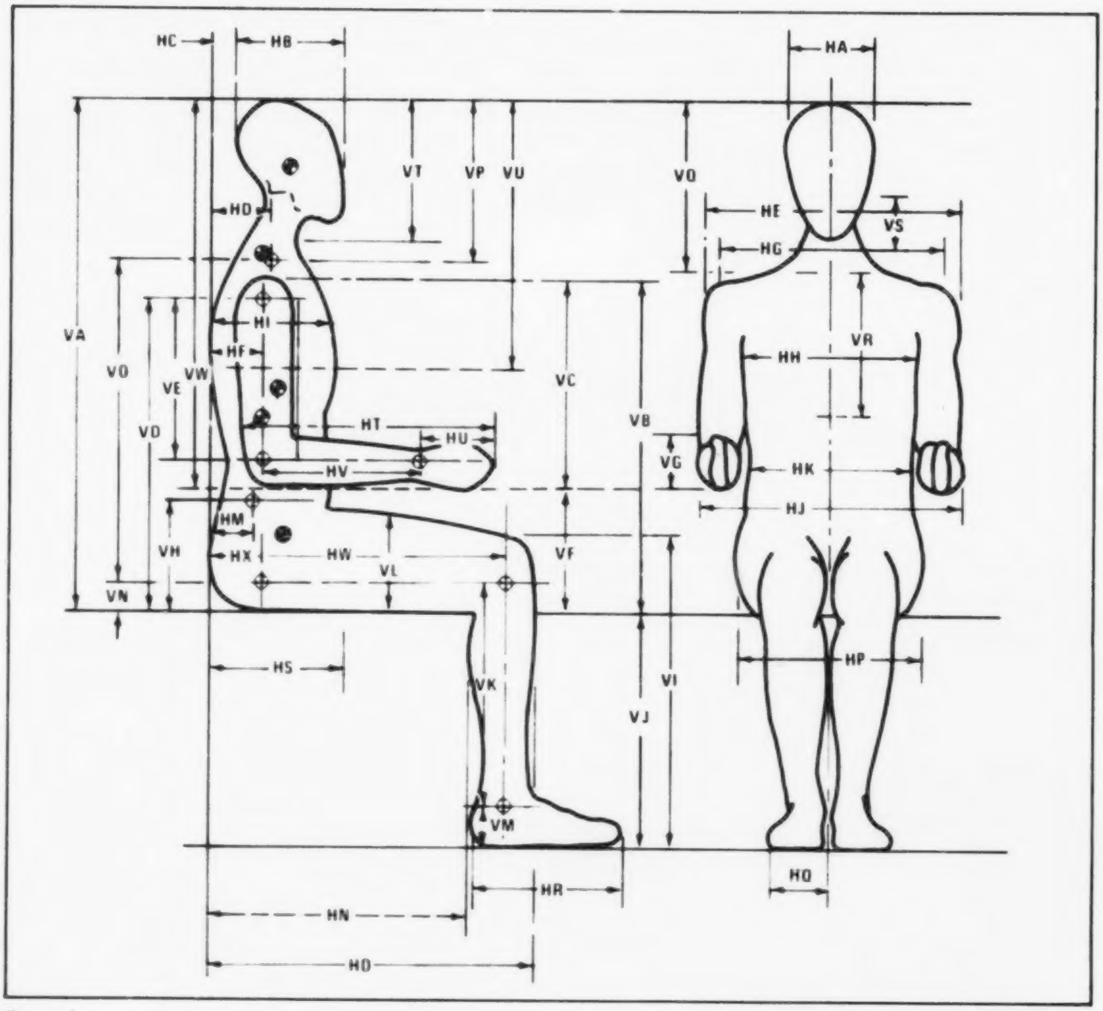

Figure 4

e. Limb joints are set at 1G, barely restraining the unsupported weight of the distal limb when it is extended horizontally. Joint preloading does not exceed 2G throughout the range of specified limb motion. Leg joints are adjusted with the torso being held in a seated, upright posture.

f. The forward edge of the lower leg rigid structure shall be a 1 inch diameter hemi cylinder set ½ inch back from the front of the knee cap.

g. An articulated representation of the clavicle shall be made, which is attached to the sternum at the supra sternal notch at their inner ends and to the shoulders at the acomium process at their outer end, to allow shoulder articulation as specified in Table 3 and illustrated in Figure 5.

INSTRUMENTATION

1. The dummy is instrumented for the measurement of orthogonal accelerations of the CG of the head and thorax, and of axial forces in the femurs and the compression of the thorax and possibly pressure rise in the abdominal sac.
2. For the purpose of accelerometer alignment, the dummy's X, Y and Z axes are located on the dummy with the dummy seated in upright posture as shown in Figure 4.
3. The accelerometers are rigidly mounted in the head on the horizontal transverse bulkhead so that the intersection of the seismic mass centers of the tri-axial cluster is in the midsagittal plane 7.5 inches above the neck-to-thorax attachment

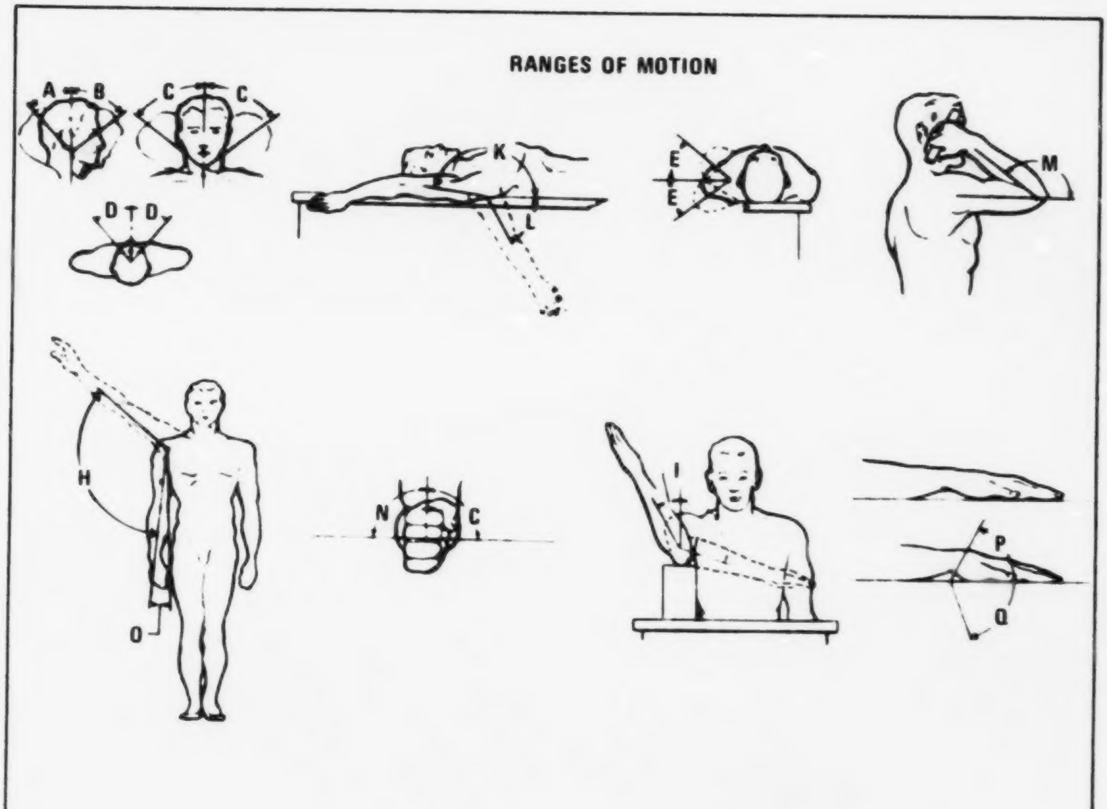
RANGES OF MOTION
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Figure 5

surface and 3.9 inches forward of the back of the head, and the X-Y-Z axes of the accelerometer cluster are parallel with the X-Y-Z axes of the dummy with maximum permissable offset and orientation misalignment not exceeding ½ inch and 5° respectively.

4. The accelerometers are mounted in the chest on a simulated spinal base so that the intersection of the seismic mass centers of the tri-axial cluster is in the midsagittal plane 5.7 inches below the neck-to-thorax attachment surface and 4.3 inches forward of the dummy's back-line and the X-Y-Z axes of the accelerometer cluster are parallel with the X-Y-Z axes of the dummy with maximum permissible offset and orientation misalignment not exceeding ½ inch and 5° respectively.
5. The femur load-sensing devices are mounted axially on each femur shaft and located 4.25 ± 0.2 inches from the knee center of rotation.
6. To preclude distortion of the test data, the resonant frequency of accelerometer mountings and attachments is at least three times the frequency range of the specified instrumentation channel.

TABLE 3 refers to Figure 5

Ranges of Motion and Terminology for 50th Percentile Adult Male O.P.A.T. Dummy

Letter Desig-nation	Title	Angle, Deg.
	Head with respect to torso	
B	Flexion	60
A	Hyperextension	60
C	Lateral flexion	± 40
D	Rotation	± 70
	Shoulder girdle with respect to torso	
E	Anterior posterior excursion	
F	Elevation	20
AG	Depression	10
	Upper arm at shoulder	
G	Adduction	0
H	Abduction	135
J	Medial rotation	90
I	Lateral rotation	0
K	Flexion	180
L	Hyperextension	60
	Forearm at elbow	
M	Flexion	135
	Hand at wrist	
Q	Palmar flexion	90
P	Dorsiflexion	60
O	Pronation	
N	Supination	180
	Thight at hip	
R	Flexion	120
S	Hyperextension	45
T	Medial rotation	50
U	Lateral rotation	50
W	Adduction	10
B	Abduction	50
	Lower leg of knee	
X	Flexion	135
	Foot at ankle	
Z	Plantar flexion	45
Y	Dorsiflexion	30
AB	Inversion	20
AA	Eversion	20
	Long axis of torso	
AC	Flexion	40
AE	Hyperextension	30
AD	Lateral flexion	35
AF	Rotation	35

THE UNITED KINGDOM TECHNICAL PRESENTATION

SECTION TWO PART 4

BRITISH LEYLAND/TRRL EXPERIMENTAL SAFETY SUBSYSTEM CONTRACTS

PETER M. FINCH, *Manager*
Vehicle Body Safety
British Leyland Motor Corporation

INTRODUCTION

Before commencing a review of British Leyland's progress to date, I would remind you that in accordance with government policy, we are not at this stage engaged in the design of an Experimental Safety Vehicle as such, but in the development of a number of separate projects concerned with specific aspects of the ESV programme (Figure 1).

BRITISH LEYLAND
UK LIMITED-TRRL

EXPERIMENTAL SAFETY
SUB-SYSTEMS

TRRL PROJECT NOs.
3 FRONTAL IMPACTS
4 SIDE IMPACTS-2 DOOR VEHICLES
5 OCCUPANT PROTECTION-HEAD IMPACTS
6 SIDE IMPACTS-4 DOOR VEHICLES-FIXED SEATS
7 OCCUPANT PROTECTION-KNEE/LEG IMPACTS
8 OCCUPANT PROTECTION-STEERING WHEEL IMPACTS

Figure 1

Moreover, we are in some areas conducting our initial investigations, using test levels somewhat different from those suggested by our American colleagues, because we believe that the vast majority of European injury-producing accidents take place at a lower speed range than do those in the United States. Also, we feel that some of the test criteria, notably the frontal impact into a rigid pole at 50 mph, set an impossable requirement for any normal production passenger car.

Except for the project on energy-absorbing steering assemblies, all the development work is being conducted on the current production 1.8 Morris Marina Saloon, a conventional front engined, rear-wheel drive, medium car having a laden weight inclusive of two dummies of 2,385 pounds (Figure 2).

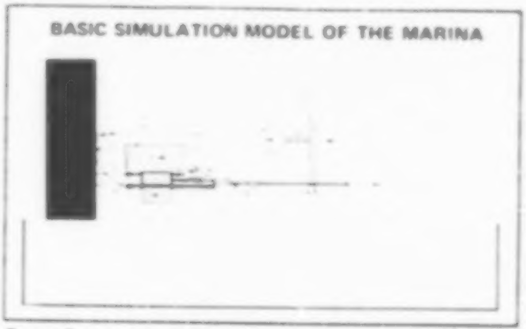

Figure 2

TRRL Project No. 3. Frontal Impacts

As with all of these projects, our first task was to conduct a series of tests on production vehicles in order to establish a base against which to assess our future programme. Tests at 10 and 15 mph were undertaken first to obtain estimates of repair costs. These were followed by 40 mph barrier tests, which increased by almost 80% the kinetic energy involved at 30 mph. It was considered that the results of these tests presented sufficient problems for commencing our Phase I development programme, when coupled with ½ offset and 30° barrier impacts.

During this initial test period, our computer simulation model was being perfected; with it we are able to include, or exclude, structural or mechanical units and vary stiffness and masses, in order to obtain predictions of the development vehicles' probable behavior under dynamic tests.

The technique basically involves the solution of a multi-mass spring system, in which the vehicle is divided into its important mass areas, connected by non-linear springs. The non-linear springs represent the major crushable components of the structure. Their stiffness characteristics are determined experimentally by controlled crushing in our crush rigs. The magnitude of the crush loads is measured by load cells attached to the barrier and the corresponding deformation measured by linear and rotary potentiometers.

For relatively simple structures, such as longitudinals or dash assemblies, end loading is sufficient, but for very complex structures, such as a complete vehicle, we endeavor to distribute the load input

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points approximate to the major mass concentrations. In this way we can obtain a close correlation between the dynamic barrier and our laboratory crush tests. By measuring the amount of work done during crushing and applying it to subsequent structural modifications, we can observe the different pattern of collapse resulting from those modifications, and thereby obtain a visual preview of their probable dynamic behavior.

Figures 3 and 4 show, first, a front end close-up of our 40 mph crash car, followed by a similar view of the crush vehicle.

No United Kingdom investigation into frontal impacts can afford to ignore the problem of pedestrians. Our traffic environment is such that a dangerous mixture of vehicles and pedestrians seems almost inevitable. Approximately 24% of all our injury-producing road accidents involve pedestrians, principally the young and the old. More significantly, they constitute 40% of all road fatalities, whereas the more publicized car-occupant fatalities represent 38% of the whole.

Figure 3

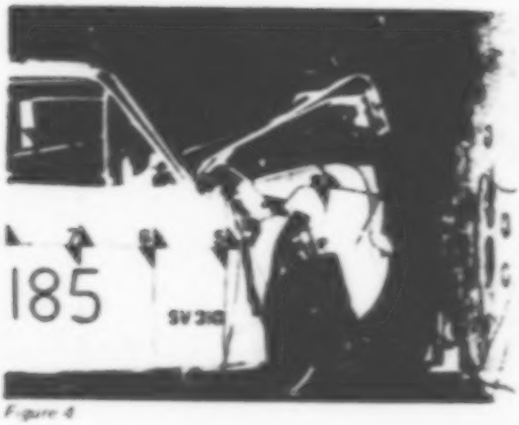

Figure 4

Regrettably, medium speed impacts are, except under the most fortunate circumstances, almost certain to prove fatal because of the preponderance of severe head injuries resulting from this type of accident. Low speed pedestrian impacts, on the other hand, should at least be capable of a reduction in injury severity, by the use of padded, energy-absorbing bumpers and grill surrounds, coupled with a suitable front-profile design. We hope to demonstrate the former on our Phase I prototypes and to investigate the latter on our Phase II.

The subject of energy-absorbing bumpers raises the problem of structural, no-damage impacts at low speeds. We are investigating, in conjunction with various specialist manufacturers, four distinct systems, the first two being velocity sensitive:

1. A hydraulic system
2. A system involving the extrusion of materials
3. A complete rubber composite bumper
4. A combined rubber buffer and collapsible energy absorbing unit

The buffer to provide for no-damage impacts of up to five mph and the bumper units or "cans" to accommodate limited-damage impacts of from five to ten mph, the resulting limited damage being restricted to these cans which should be easily replaceable.

As with all developments there are a number of problems still to be solved and designs assessed before arriving at the most reliable and cost effective system. Complementary to whichever bumper systems are chosen, the Phase I prototypes will incorporate a soft grille and headlamp surround, so as to extend the no-damage frontal area as much as possible.

At this moment, the Phase I design for frontal impacts is almost complete. Figure 5 shows the lower structural design which outlines the major structural modifications envisaged for the first phase prototypes.

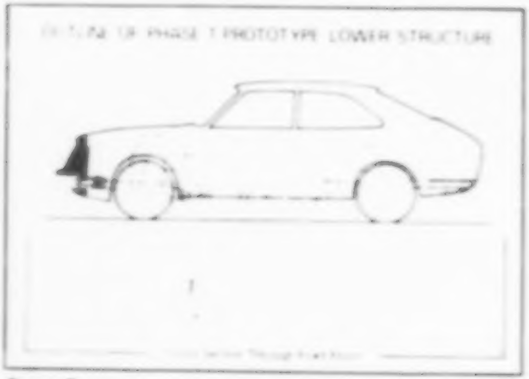
OUTLINE OF PHASE I PROTOTYPE LOWER STRUCTURE
Section Through Front Floor

Figure 5

For the purpose of this paper, TRRL Projects 4 and 6, dealing with Side Impacts, will be considered together, as will Projects 5 and 7, which we associate with Occupant Protection.

TRRL Project No's 4 and 6 – Side Impacts

With these projects the problem is twofold.

Firstly, to prevent crushing injuries through engineering, a vehicle structure capable of resisting undue intrusion from a variety of side impacts, and secondly, to provide a structure with an internal, energy-absorbing system able to reduce the acceleration forces induced by the occupant's impact with the interior.

Project 4 is aimed at the improvement of two-door saloons, while Project 6 is concerned with the performance of four-door saloons. As with the former project, our first task was to establish the performance of our current production vehicles by a series of impact tests. These included

1. car-to-car and mobile barrier-to-car up to 30 mph
2. car-to-car at 45° and at 20 mph
3. car-to-pole tests at 15 mph

The mobile barrier tests were included to provide a comparison of this type of impact with the 90° car-to-car tests. As expected, because of the absolute rigidity of the mobile barrier face, as compared to the collapsible, energy-absorbing nature of the bullet car, the resultant damage was somewhat different, producing slightly less penetration from the mobile barrier, but a greater extension of the damage fore and aft (Figure 6).

Structurally, the car-to-car impact at 45°, and the car-to-pole impacts, probably present less of a problem than the 90° car-to-car impacts, although some of the recorded dummy accelerations were very high.

The performance criteria of all these projects is not restricted to the degree of passenger compartment integrity, but also considers the occupant acceleration and load tolerance levels.

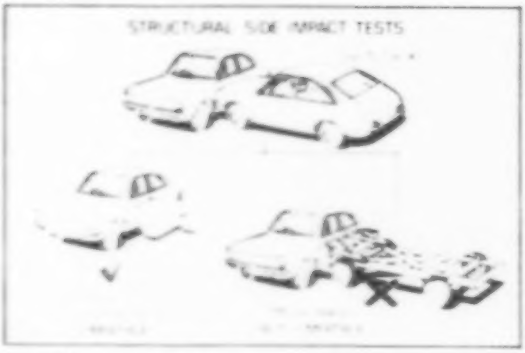

Figure 6

Human tolerance data, particularly on side impacts, still require considerable research; scientific knowledge of what is, and what is not, acceptable is very limited. The biomechanical behaviour of dummies under tests also leaves much to be desired.

However, although we hope that such information will ultimately become more realistic, we are endeavoring to obtain some measure of the impact accelerations and forces produced on the occupant by means of a dummy specially developed by TRRL for this purpose. With this dummy we can measure the shoulder, rib, iliac crest and pelvic loads, in addition to the head, thorax and pelvic accelerations.

Once the production vehicles had been tested, the next problem was to obtain the stiffness values of the main structure, and its component parts, for inclusion in our side-impact computer programme. Modifications to the components, by means of structural design or plastic foam filling, have been physically measured and their overall effect upon the complete structure is being assessed by means of the computer simulation.

During the initial development period we found we were able to reproduce the structural damage, caused by a 90° car-to-car impact, by fitting a solid ram to the face of one of our crush rigs. The ram profile being taken from the deformed front structure of the bullet car. Figure 7 shows the resultant deformation of the crushed vehicle.

Improvements in the design and energy-absorbing characteristics of the internal trim are, meanwhile, being assessed by means of impacts with various body forms, each containing means of measuring accelerations, penetration and rebound.

The two-door version, with its tip-up front seats, necessarily offers a cheaper, but probably less effective, answer to the problem than does the four-door. Such a vehicle design must have a greater potential

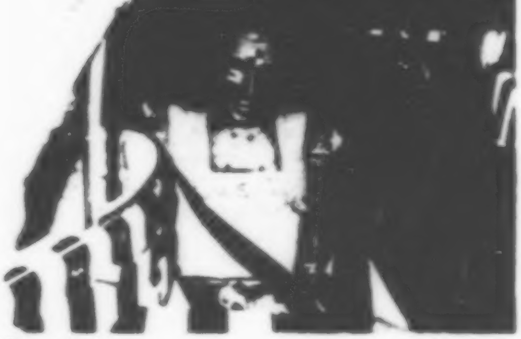

Figure 7

for improved performance, since the front seats can be fixed, or made integral, with the basic structure, thus providing a bulkhead linking both vehicle side structures together. This approach offers an added advantage in ensuring optimum locations for all sizes of occupants, relative to the restraint system provided (Figure 8).

For example, seat belt anchorage location problems associated with ultra tall, or ultra short, occupants may be overcome by incorporating a seat belt system into a seat made sufficiently rigid to accommodate the required load (Figure 9).

The integral seat approach does, however, necessitate the incorporation of a fully adjustable steering wheel and pedals, as well as requiring the location of all other controls within the reach of the shortest driver (Figure 10).

A road-proving vehicle, incorporating such a system, and including the new door trim and facia layout is currently under appraisal.

Side-impact tests of the Phase I two and four door prototypes are due to commence at the end of this month.

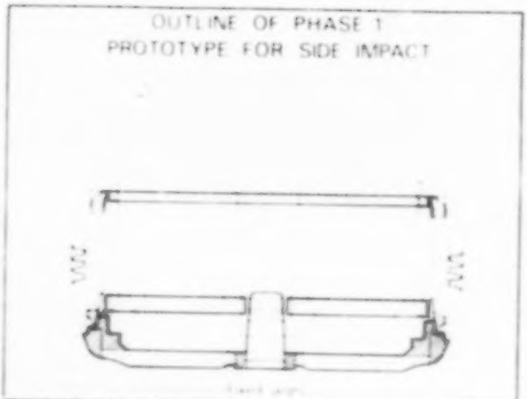

Figure 8

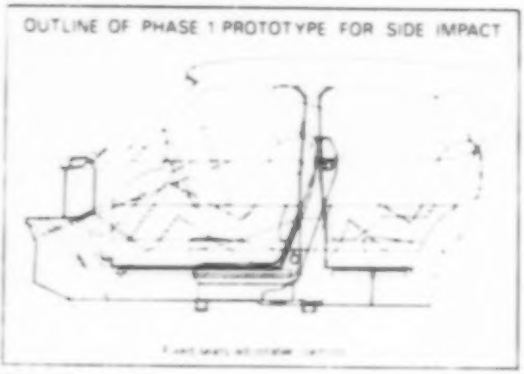

Figure 9

Project No's 5 and 7 – Occupant Protection

Basically, Project No. 5 encompasses occupant protection above the waistline of the car, while Project No. 7 is concerned with those areas below the waistline.

As with the previous projects, our first task was to establish the impact performance of current production vehicles. These we are endeavoring to obtain by a variety of test procedures, dependant upon the complexity of the problem and the relevance and repeatability of the test. For example, in addition to the anthropomorphic dummy measurements obtained from the dynamic crush tests already described, other basic criteria are being established from sled tests and from a variety of head, shoulder, hip and knee impactors. The latter enable us to assess force-distributing materials and internal structural supports more accurately than by using dummies, once the directions and points of impact have been established.

The lower windscreen area is being redesigned to accommodate the glass retention rubbers well below the level of the facia, in order to eliminate facial lacerations caused by impact with those fragments of glass which normally remain supported in the glazing strip. A suitable sled test incorporating a dummy with its head covered with a layer of foam and chamois leather provides a measure of probable facial damage.

The design of the interior occupant protection system is being aided by simple computer simulation programmes and it is hoped to develop this approach still further in the near future. Meanwhile, a prototype incorporating Project 6 modifications is being prepared for occupant acceptability trials.

Project No. 8 – Steering Assembly Impacts

This project is concerned with improved, energy-absorbing properties of both high rake and low rake

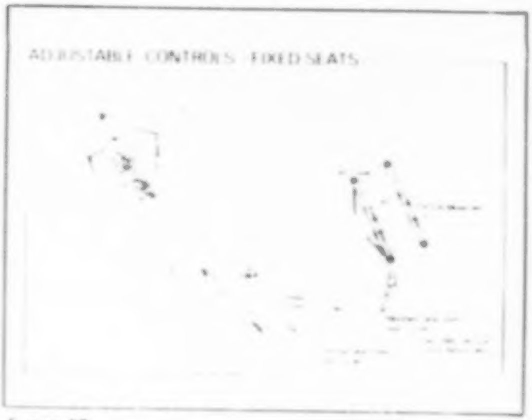

Figure 10

steering assemblies. We have only recently commenced work on the problem, although tests on existing equipment are under way and a simple computer programme has been developed. The MIRA/Ogle dummy is being used for these investigations, as we consider it offers the best simulation of the thorax/head complex.

GENERAL COMMENT

In concluding the progress report of the British Leyland/TRRL experimental safety subsystem projects, I would emphasize that, although the individual contracts are being programmed and presented as entirely separate entities, it is also our ultimate intention to combine these projects into a number of total assessment vehicles and, where feasible, this is being taken into account during the design stage.

Figure 11 indicates in a generalized form, our progress to date. As you can see, we expect to be able to present you with details of our Phase I prototype tests at the next ESV conference in London.

We are also planning to show you one of our composite vehicles and outline our progress towards the completion of Phase II.

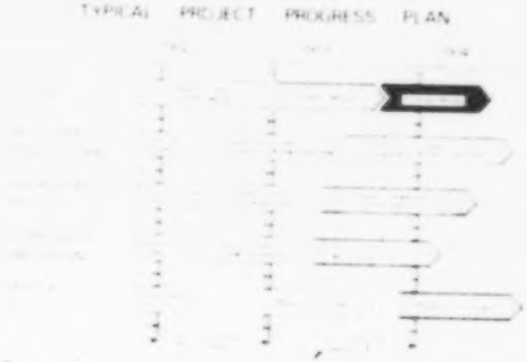

Figure 11

As a postscript, our Austin Morris Assembly Division at Longbridge is currently discussing with TRRL an additional series of projects involving the development of three transverse engined models, namely the Mini, the 1,100 and 1,800 saloons. Testing will include, in addition to the ESV impacts, a selection of mixed car-to-car impacts aimed at minimizing aggressiveness.

THE UNITED KINGDOM TECHNICAL PRESENTATION

SECTION TWO PART 4

A REVIEW OF SEAT BELT EFFECTIVENESS AND INVESTIGATION OF POTENTIAL IMPROVEMENTS, INCLUDING PASSIVE SYSTEMS

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Presented by
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INTRODUCTION

It is now almost universally accepted that the use of a seat belt confers on the motor car occupant an improved degree of safety in a vehicle accident. Without, at this time, entering into any controversial discussion on the subject of alternative vehicle safety systems, this paper is intended to outline work progressing in conjunction with the Transport and Road Research Laboratory.

1. The extent to which, in real life, with the existing types of vehicles, road conditions and traffic situations, the seat belt meets the requirements for occupant protection at an economic cost.

2. The potential for improvement by research and development work in the performance of orthodox, three-point seat belts to make them more effective in most accident situations.

3. The design of reasonably priced passive systems for installation on cars going into production, that would be acceptable to the user both aesthetically and in practice.

ANALYSIS OF ACCIDENT AND INJURY DATA

The assessment of a product as used by the customer, rather than as tested by engineers in a laboratory, is always very important. It is particularly necessary in the case of car occupant restraint systems because of the difficulties associated with reproducing conditions in the laboratory which are representative of 'real life' crashes, and of making dummies which correspond to the human body. In addition to these problems, there are difficulties in assessing laboratory tests, because of a lack of knowledge of human tolerance levels, and in reproducing such real life variables as the amount of clothing worn by car occupants. The non-ideal way in which a seat belt is often worn, or the way in which the occupant may be sitting, can also result in variations which are difficult to evaluate.

A study of the overall effect on injury levels resulting from the use of seat belts has been performed by the TRRL (Reference 1). This study concludes that seat belts with upper torso restraints reduce the chance of a car occupant receiving a serious injury by a half. The result agrees with that obtained by other authors, in particular N. I. Bohlin, (Reference 2) quotes an effectiveness of between 48% and 63%.

THE STUDY OF INJURIES RECEIVED BY RESTRAINED OCCUPANTS

In order to further improve the effectiveness of seat belts, it is first necessary to study how the existing systems perform and to analyze their limitations. This has been done by studying a collection of over one thousand very detailed accident reports which have been prepared by TRRL personnel. The reports cover accidents which have occurred in the area shown in Figure 1.

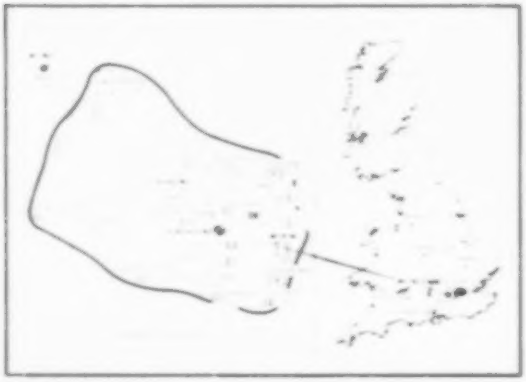

Figure 1

The reports describe the site of the accident, the probable paths of the vehicles prior to and after impact, the damage sustained by the vehicles involved, the injuries received by the occupants and the mechanism by which each of these injuries was

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received. Cases for these reports were obtained by interviewing persons who were detained in the hospital. These interviews were carried out by medically qualified personnel. Because of this method of data collection, all levels of injury were included but the more severe cases tended to predominate. The sample used in this study included 911 unrestrained occupants receiving injuries more "severe than minor" (for classification of injuries, see Reference 3), there were 71 cases of persons using three-point belts and seven cases where other belt systems were used which resulted in similar levels of injury.

In the 71 cases in which a restrained occupant was injured, the location and severity of the injuries were studied. The frequency of injuries to various parts of the body was then calculated and compared with similar figures taken from the 911 cases of unrestrained occupants. These results are shown in Table 1 below.

Table 1

In column one is shown the frequency with which each area of the body has received an injury more severe than minor for the case of unrestrained occupants. In column two are the same figures for the sample of occupants restrained by three-point belts expressed as a percentage of the number of restrained occupants at risk, that is, the number of persons who would have been injured if they had been unrestrained. The method used to calculate the number of persons at risk is shown in the Appendix. In the third column is the reduction in injuries which can be attributed to the use of seat belts.

Discussion

It can be shown that the results for the abdomen and the hip pelvis are not significant statistically and do not, therefore, indicate any effect due to belt usage. In the same way the results for the head, chest and lower limbs have been found to be significant at the 1% level and the reduction of these injury levels should not be attributed to chance.

In an attempt to gain a better understanding of the mechanism of the injuries received by restrained occupants, the part of the car with which each injury was associated was considered. It soon became clear that in rear and side impacts, the part of the vehicle struck by the occupant was rather random and not greatly affected by the presence of the belt. For the purpose of defining the mechanism of injury, the sample of accidents was, therefore, limited to include only impacts with some frontal component. Minor injuries were also included in this analysis, because although these injuries were not in themselves important they did indicate a point impact between car and occupant. Each of the most common areas of injury was considered for the effect of seat belt usage and the cause of the injuries which were not prevented by the use of a three-point belt.

Head, Face And Neck. Table 1 shows a 56% reduction in the incidence of head and face injuries resulting from the use of belts. Only one case of deceleration injury was seen without evidence of the head having struck the interior of the car and there is some doubt about the cause of this injury. Only one neck injury, other than slight stiffness, was noted despite the fact that none of the cars was fitted with head restraints. Insufficient restraint or excessive intrusion which allowed the head to contact the steering wheel was responsible for over half the injuries, the remainder resulted from contact with either screen, header, bonnet, roof, 'B' post or door window frame.

Chest. A very large reduction in chest injuries was noted, if the belt-induced injuries are excluded from this section (these were often of a moderate or minor type and are dealt with below under a separate heading). The remaining chest injuries resulted from either passenger compartment intrusion or the belt providing insufficient restraint, which allowed impacts with the steering wheel, facia, and 'A' post.

Abdomen. Considering injuries other than belt induced, the apparent increase in this area has been shown to be statistically not significant, due to the small size of the sample. The injuries were the result of impacts, with either the steering wheel or the facia, due to insufficient restraint or excessive intrusion.

Pelvis, Hip And Lower Limbs. Injuries in these areas are those most commonly received by the restrained occupant. In particular, the injuries to the pelvis and hip joint are not significantly altered by the use of a belt, which is particularly disturbing because of the probably long recovery period or permanent consequences. In the frontal impacts, 23 cases of injury to the hip and to the lower limbs resulted from the facia and parcel tray being struck, due to insufficient restraint of the lower part of the body. However, in 13 of these cases major passenger compartment deformation was a contributory factor. In seven cases the door was struck, often after it had been deformed and in another five cases the steering wheel, column, gear lever or wheel arch were impacted.

Belt Induced Injuries. — Although seat belts were very effective in preventing serious injury, cases were found in which the occupant received minor injury as a direct result of the loads imposed by the belt. In 28 cases the injury was limited to either bruising or to the fracture of a few ribs. Ten cases were recorded in which the chest was more seriously damaged, i.e. the clavicle or sternum was fractured, often together with a few ribs. Although very unpleasant, these injuries are not usually considered very serious.

Eight cases of serious injury were recorded. In two of these, abdominal injuries were due to the buckle being worn near the center of the body and in four cases the occupant was subjected to additional loads from the rear due to the seat being pushed forward by vehicle deformation or unrestrained rear seat occupants.

Conclusions from the Study of Seat Belts in Service

Seat belts have been shown to be very effective in preventing a large proportion of the injuries which are received by the occupants of motor vehicles. They are particularly effective in reducing the incidence of head and chest injuries. The effectiveness of the belt in preventing lower limb injuries, and particularly those to the hip joint and pelvis, is not high. The lack of restraining effect on the torso is, in part, due to the slackness with which belts are often worn. Even if better user education or retractor reels could eliminate this factor, obesity, bulky clothing, webbing stretch and 'layout slack' (Reference 4), can also result in excessive occupant movement. In order to prevent the injuries which result from this, it will be necessary to either provide more space, a suitable collapsible knee restraint or increase the restraining effect of the belt by the use of crash-deployed belt tensioners.

The number of head injuries resulting from excessive forward movement and the number of belt-induced chest injuries are large. The next section describes methods that have been developed to reduce the belt tension in the upper torso restraint and at the same time control head movements so as to prevent the driver's head from contacting the steering wheel.

IMPROVEMENTS IN SEAT BELT PERFORMANCE

Objectives of Development Programme

Previous theoretical studies into the dynamics of restraint systems and the mechanisms of occupant injury have demonstrated certain important principles. Investigations were commenced on the basis that occupant injury was related mainly to the peak deceleration imposed and that existing orthodox seat belt webbing was a poor energy absorbing material imposing considerable rebound effect during restraint (Reference 5).

The purpose of the development programme was, therefore, to design a seat belt system that would permit controlled forward movement of the occupant inside the vehicle during an accident, in such a manner that loads and deceleration levels applied to the body were minimized. The required performance was unlikely to be achieved with existing seat-belt webbing, therefore, the use of energy absorbing load limiting devices was considered necessary.

Laboratory Test Work to Optimize Belt Performance

The test rig used for these investigations incorporated two strongly constructed car seats mounted on a superstructure with appropriate seat-belt anchorage points. This, in turn, was mounted on a sled at the MIRA impact-test facility where all the laboratory tests were carried out. The impact speed selected was 30 mph, all the tests were frontal impacts and the deceleration characteristics were selected as an approximation of several typical British saloon cars. Alderson 50th percentile adult male dummies were used in these tests.

Some criteria were required to provide a comparative means of assessing test results, and when testing commenced, it was considered most appropriate to use the following:

1. *Head.* A "Gadd Severity Index" limit of 1,000.
2. *Chest.* A resultant deceleration of the chest not exceeding 60g for more than 3 m.s.
3. *Femur.* A compressive load on the femur not exceeding 1,400 pound-feet.

Early tests soon indicated that the requirement for the head could not be achieved using conventional lap and diagonal seat belts. This first series of tests confirmed high-head decelerations, in some cases exceeding 300 g, and high-belt load measurements, peak values in the shoulder belts reaching 2,500 pound-feet.

A second series of tests showed that a substantial reduction in Gadd severity-index values to less than 1,000 could be achieved by incorporating an energy absorber in the shoulder section of the seat belt. One of the energy absorbers used extensively in these tests was ply-tear webbing. This webbing dissipates energy by tearing apart two sections of webbings woven back to back. The material begins to tear at a predetermined load and then continues progressively with a substantially constant load of a somewhat lower value. It was also observed that the dummy rebound was much reduced as a result of the energy dissipation, thus confirming the original theoretical predictions. Measurements of load in the belts indicated an improvement with the peak shoulder belt load reduced to 1,400 pound-feet.

One of the limitations in the use of energy absorbers was the space available within the passenger compartment of the vehicle, since, if the increased forward movement permitted the occupant's head to strike the vehicle interior, the benefit derived from reduction in head deceleration would be lost. Some of the second series of tests incorporating energy absorbers achieved low Gadd severity indices, but resulted in excessive movement when related to the space available in the majority of British cars.

To retain the advantages of energy absorbers, while ensuring a reduction in the forward movement, a third series of tests were undertaken in which ultra-low elongation webbing was used together with energy absorbers, principally, tear webbing. Figure 2 illustrates some of the test results from each series, the target-performance area being that which combines a head-severity index of below 1,000, with forward movements low enough to avoid occupant contact with the vehicle interior.

Figure 2

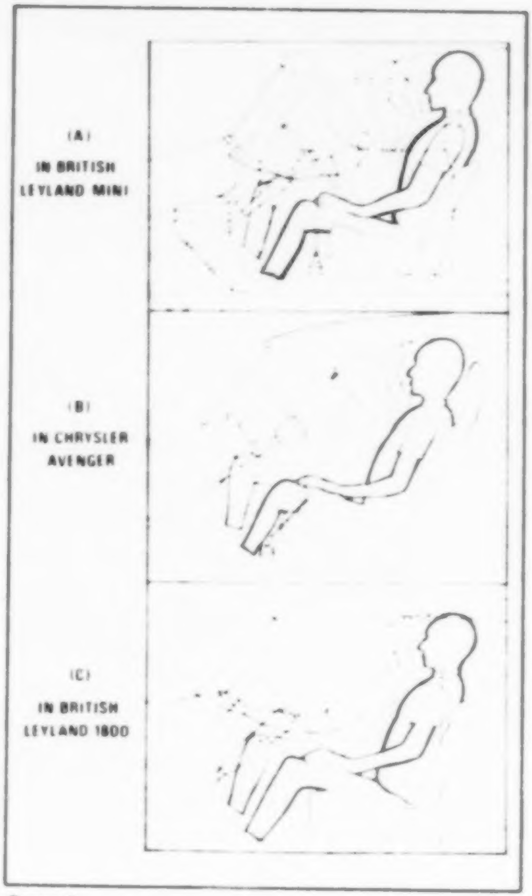

Figure 3

Figure 3 shows occupant movement related to interior space available for three typical British cars. The results were achieved with 4% elongation patent webbing and a 1,500 pound-feet ply-tear webbing. The trajectories were for a 50th percentile adult male with the seat in the mid position. It can be seen that no contact with the vehicle interior occurred, but when the seat was in the fully-forward position contact with the steering wheel was indicated on one vehicle, with an anticipated increase in the head injury criteria value. This point emphasizes the need to tailor the belt system to suit a particular vehicle by use of a different rating-energy absorber to prevent contact. In some cases, incorporation of a yielding or well padded steering column will have to be considered to prevent high-local loads on the occupant's face and chest.

All the test work described has been based on 30 mph frontal barrier impacts and the problems of higher speeds must be considered. The forward movements would be greater and while this could be accommodated by use of higher-rate energy absorbers, such a system would impose undesirably higher loads and decelerations in lower speed impacts. A more acceptable solution might be to optimize the seat belt installation for a specific barrier impact speed and supplement this with interior padding and a collapsible steering column for higher speeds.

It can be seen from the illustrations in Figure 3 that forward rotation of the head relative to the torso occurs on impact. This is undesirable insofar as it results in increased forward movement and high angular and linear accelerations. It has been found that this effect can be reduced by allowing the lower part of the body to move forward and downward at the early stages of the crash. This applies a counteracting force to the head, so that the resulting forward rotation is limited. This controlled forward torso movement can be effected by use of energy absorbers in the lap section of the seat belt, but additional space in front of the knees may be required in certain vehicles.

Conclusions From Laboratory Tests

Improved protection can be obtained from a seat belt system by careful design to minimize the loads and decelerations experienced by the occupant.

Tailoring the belt system to suit the particular vehicle is necessary if the optimum performance is required, since vehicle interiors vary greatly in space and shape.

Because of the presence of the steering wheel, alterations to some vehicles may be required to ensure the best possible driver protection.

PASSIVE SEAT BELT SYSTEMS

During the last two years, complete, fully operational passive seat belt systems have been installed on five different motor cars and experimental schemes have been built on four vehicles. This work has served to provide a good basis for consideration of all aspects, from vehicle design to ultimate user. Not least of the problems is the conflict between the ideal, which may be complex and expensive, and the practical, which must be cost effective, reliable, and acceptable to the final user. Many different types of passive seat belt systems have emerged as a result of development work, and it is possible that certain systems may be adapted more easily to particular motor car designs. One specific system installed in a medium size British car, will be described to illustrate design considerations.

1500 c.c. Saloon, With Cord Actuated System

This vehicle was first equipped with an experimental version of the passive seat belt system over twelve months ago, and since then has been modified in the light of experience, until it represents a scheme that could be readily applicable to a wide variety of car designs. It is convenient in operation, and should not be expensive if designed into the vehicle as original equipment.

The seat belt system as shown in Figure 4 is basically a three-point layout with the diagonal and lap straps mounted outboard to the rear edges of the car doors and the reels mounted inboard adjacent to the transmission tunnel. To ensure adequate strength, the upper door mounted anchorage is arranged to latch into the "B" post so that any imposed load is transferred from the door frame to the pillar. An additional feature is that the shoulder belt anchorage-point height is instantly adjustable. In the context of improved belt effectiveness this is considered important to ensure optimum conditions for varying sizes of occupant, particularly children.

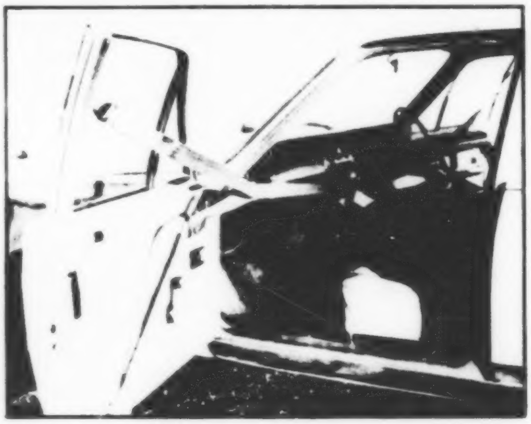

Figure 4

Retraction and deployment of the belts is effected by use of cords attached to webbing guide loops arranged to pull the belts forward, well clear of the occupant. The guide loop controlling the inboard section of belt pulls directly, while the outboard webbing guide is located in a track running on the inside of the door. The power to operate this system is supplied through an electric motor, gear box and pulley unit, one of which is required for each of the front seat belt installations. Differential pulley diameters are used to suit the varying movements required by each section of the webbing, and the cord operating the slide in the door is conveyed in a guide tube, through the door-hinge axis, to the motor unit, which is mounted under the bonnet. Channels are provided in the tunnel and center console area so that the actuating cords disappear from view when the seat belts are in the deployed position around the occupant.

The electrical circuit diagram shown in Figure 5 illustrates the control system used on this particular installation. in principal, the opening and closing of the door signals the actuation of the seat belt system, but there are other important requirements to ensure satisfactory operation. The normal type of courtesy light door switch is unsatisfactory for actuating the mechanism and has been replaced by a switch on the "B" post that signals immediately when the door is opened and not again until it is firmly closed. For added user convenience, push buttons are included so that the seat belts can be retracted clear of the occupants without opening the doors. A switch on the hand brake, will automatically redeploy the belts if they are left forward when the car is driven away.

Figure 5

An important aspect of the installation is safeguards against interference, either accidental or willful. With this in mind, "loose cord" switches are incorporated so that if the belts should in any way be prevented from deploying fully around the occupant, the motor unit will switch off, but will complete its operation when the obstruction is removed. At this point, it should be stressed that if the actuating cords are cut or broken, the normal retraction spring in the reel will bring the belts into the driving position, around the occupants. One further control feature is the inclusion of a seat switch on the driver's side so that, when the door is closed, the belts are only deployed if there is an occupant. The benefits of this are obvious in terms of the reduction in frequency of operation of the mechanism and added convenience when re-entering the vehicle.

Proving Tests and User Evaluation

Two vital aspects of this work are laboratory proving tests to ensure reliability of component parts of the system, and user evaluation trials to assess the degree of acceptability by the general public.

Figure 6 shows a complete seat belt system with reel and motorized pulley unit, which is undergoing endurance tests in a range of environmental conditions to ensure satisfactory operation. All parts of the systems are being similarly subjected to laboratory proving tests.

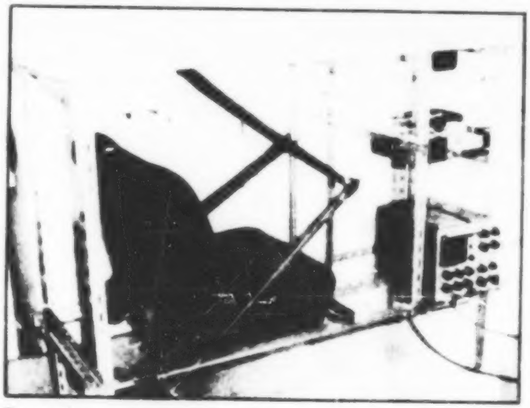
Figure 6

The TRRL has commissioned Loughborough University to carry out an independent investigation into this passive seat belt system, with respect to user evaluation. This study is now under way, and it is planned that the reactions of a large number of people will be assessed by a team of psychologists, and ergonomic specialists.

CONCLUSIONS

It has been confirmed that seat belts have been effective in preventing a large proportion of the injuries which would otherwise have been received by vehicle occupants in accidents. Accordingly, it is expected that the U.K. requirements will continue to be met by seat belt systems. These systems may well incorporate improvements which the research work has investigated and found to be both practical and potentially beneficial. Passive seat belt systems have been extensively evaluated in several vehicles and useful experience has been gained. A formal user reaction survey is in progress, results from which will be available shortly, however, reactions to date have been predominantly favorable.

Regarding future research, a further series of sled tests using car bodies with alternative seat belt systems will be undertaken with impacts at a range of speeds and angles. Arrangements have also been made to carry out crash and roll-over tests on cars equipped with passive seat belt systems. Investigation is furthermore considered desirable into the subject of automatic seat belt tightening on impact.

APPENDIX

The Comparison of the Injury Rates for Restrained and Unrestrained Occupants

In order to assess the performance of seat belts it is necessary to compare the rates at which restrained persons receive various injuries with those for unrestrained occupants. When the sample is selected using a criterion of a certain injury level, occupants who are prevented from receiving such an injury, by virtue of wearing a belt, will be excluded from the evaluation. A comparison based on only injured persons would result in a low value for the effectiveness of seat belts. It is, therefore, necessary to apply a correction to the figure for the number of restrained persons injured so that the frequencies of injuries are calculated in terms of the number of persons "at risk." In the case being considered in this paper, 71 restrained occupants were injured. Calculations based on observations of seat belt usage in the area and period covered by the sample, indicated that restrained occupants were underrepresented by a factor of about one third. This results in the number of persons "at risk" being 106 and it is on this figure that the injury rates in column two of Table 1 have been calculated.

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THE UNITED KINGDOM TECHNICAL PRESENTATION

SECTION TWO

PART 4

DEVELOPMENT AND IMPROVEMENT OF IMPACT TEST METHODS

DR. M. A. MACAULEY, *Head of Research*
Motor Industries Research Association

We do not make vehicles, but we carry out tests for vehicle makers and also do background research on the performance of these tests. I shall not comment on developments in vehicles, but only about impact-test methods and the relevance of these methods to the problems they are intended to solve. Vehicle impact testing on a large scale to legal requirements is only a few years old and although the severity and complexity of the tests have been increasing steadily this has been as a continuous extension of the original requirements, rather than as a reevaluation of these in the light of experience gained in service. Such a reassessment will need to be carried out before long and this paper is intended as a contribution toward it.

Testing is always a compromise between what is desirable and what is feasible. An ideal test procedure takes only the essential features of a real-life situation and incorporates these in a simple and repeatable test. In practice, the requirements of realism and simplicity conflict. Sometimes they conflict badly and then, unless we are very careful, a dangerous situation arises in which the tester has to choose between doing something obviously stupid, which will get him through the test, or doing something obviously sensible which will fail him. Even if we surmount this, there is still a further problem because it is perfectly possible to have a good test procedure, ensure that it is complied with sensibly and still do more harm than good in the real-life situation.

None of this is an excuse for not trying but it does show that it is necessary to check continuously on whether the tests being carried out really improve the real-life situation. This is not unique to vehicle impact testing. It is always difficult to decide which are the essential aspects of a complex situation, so that the assumptions underlying test procedures are often in error. The more complex the situation the less likely we are to getting things just right in the first one or two attempts.

There is no doubt that we have all become much better at carrying out impact tests to the specified requirements. We, ourselves, now do more tests in a month than we used to do in a year five years ago and the rate is still going up. The tests are also much more reliable and much more closely controlled, despite a considerable increase in complexity. Despite this, there are still drawbacks to the tests, as tests, regardless of their relevance to the real-life situation. The main one is that they are so unrepeatable that they are extremely difficult to interpret.

This is due, at least partly, to poor definition of the test requirements. The impact conditions for the vehicle itself have been reduced, in most cases, to a highly-simplified and stylized test, which is simple enough to give repeatable results and yet realistic enough to be acceptable to most people. The requirements for the dummy occupants have not been clarified to anything like the same extent. As a result, the specification of these dummies is so loose in some respects that perfectly legitimate variations in dummy construction can have larger effects than any practicable modification to the vehicle or any foreseeable change in the criteria for passing the tests. In other respects, the specification is so rigid that we are hampered in using improved dummies, even though we know areas in which the present specification is irrational and misleading.

A further reason for lack of repeatability is the large number of variables which are left to the tester's discretion. Vehicle makers have known for a long time that it is impossible to get repeatable results, even when checking internal dimensions with a simple, static mannikin unless a clearly defined, standard setting-up procedure is used. With the much more complex impact dummies such a standard procedure seems even more necessary, but existing procedures are inadequate and unrealistic. This is particularly important as it seems very unlikely that dummy occupants will ever be simplified to the same extent as the vehicle impacts.

Let us now look at the next aspect of the problem of testing. How realistic is the test situation? I have chosen, almost at random, four examples to illustrate what I think are the four basic problems in getting a

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good test procedure. The first is that it may not be possible to get a satisfactory test to simulate a very complex situation. A case in point is vehicle rollover. I said earlier that, in most cases, the simplified impact of the vehicle itself is sufficiently realistic to be acceptable to most people. The basic reason is that the overall dynamics of the vehicle in real life are pretty simple. In rollover, the overall dynamics are much more complicated and give much more scope for variations from test to test even in nominally identical conditions. As a result, tests which are sufficiently simplified to give repeatable results are misleading, tests which are sufficiently complex to give realistic results are not repeatable and the best compromise available to date is neither a very good test nor a very good simulation. Nonetheless, it may be the best that can be done.

The second problem is that in some cases we do not know what we are trying to simulate. A case in point is the neck of the dummy occupant. We know that seat belts work well in service; we know that in impact tests using seat belts, dummy head decelerations are often higher than the regulations allow; we know that one of the main factors influencing these decelerations is the dummy's neck; we know how to alter the necks to get the dummies through the test. What we do not know is what neck simulation we should be aiming at and how relevant the head decelerations are to the situation anyway.

The third problem is that of misinterpreting the available evidence. It is inevitable that this will happen from time to time, especially in a new area of activity which covers a number of existing disciplines so that no one is an expert. A case in point is the chest of the dummy occupant. Test results exist showing the behaviour of the human chest on impact. Originally, the dynamics of the chest on impact were misunderstood but I believe that they are now much better understood, even though much work remains to be done. Test procedures were defined on the basis of the original misunderstanding and there they remain so that we are not allowed to make use of our improved knowledge.

The fourth problem is that there can be a divergence between passing the test and common sense. For example, one of the criteria for passing the test is the deceleration of the dummy's head. In some cases, it is possible to get lower decelerations by letting the dummy strike its head on the steering wheel rim, but in real life this would increase the damage to the driver.

Let us now turn to the final aspect of testing. Is all this activity improving the behaviour of vehicles in real accidents? So far, I think we must say that we hope so, but we are not quite sure and we shall not know for some time to come. We are all learning fast on this aspect, as well, but it takes time for evidence to accumulate, then it has to be analyzed and finally we have to decide what it means. In a complex situation, this is seldom easy because so many things are inter-related and there are a few simple and obvious answers. For example, in pre-legislation vehicles drivers injure their chests on the steering wheels. In post-legislation vehicles the steering wheels and columns do not move back toward the driver on impact and they collapse under load. What appears to be coming out of current accident studies is that chest injuries are reduced as was expected. But when an unrestrained driver does not hit his chest hard on the steering wheel he has to get rid of his kinetic energy in some other way and the accident studies indicate an increase in the severity of other injuries. We may still turn out to have improved the overall situation but it is not so clear cut as it looked to begin with.

I should like to end on a more cheerful note. I have dwelt on problems rather than achievements because I believe that the problems are urgent and need to be aired. It would have been more pleasant to present a paper on our achievements, which have been considerable, even though they all began as a list of awkward problems, rather like the ones I have just outlined. I hope that we shall overcome our present problems equally successfully but we shall not do this by pretending that they are not there.

The dummy occupant is one of the most important items in a crash test and we have been working on dummy improvement for several years under a series of contracts from our government and from industry. There is not yet enough background information on real people for anyone to make a completely satisfactory dummy, but I believe that we have provided better interpretation of the existing data than has been done before. At present we are working on the neck, which is the problem of major interest at the moment, under a joint contract from the Transport and Road Research Laboratory and some of the major motor firms. I think this is a good combination. The vehicle makers obviously want a neck which does not make the test requirements unrealistically severe. We want a neck which will be easy to use as a test device and the government wants a neck which will give good correlation with real-life accident results. This is a tall order and so far we have not had any brilliant ideas on how to satisfy all three demands, but we are learning a great deal about dummies' necks and we hope to publish the results when the work is completed.

THE UNITED STATES TECHNICAL PRESENTATION

SECTION TWO PART 5

INTRODUCTION

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Office of Assistant Secretary for
Systems Development and Technology
Office of the Secretary of Transportation

The presentations this morning will cover the ESV Test Program which will be given by Norman Stahl of the ESV Program Office and Fred Arndt of Dynamic Science, the test contractor. This will be followed by Louis C. Lundstrom of the General Motors Corporation who will discuss "Conclusions and Projections on ESV Development." The Ford Motor Company's presentation is given this morning by J. D. Collins and K. H. Arning, and will address the results of the Ford ESV development. Results of the "Optimized ESV Integration and Tradeoff Studies" now being conducted for the Department by AMF (Incorporated) will be presented by Alan Roth, William Rap and William Wingenbach. Finally, William Scott, Acting Director of the ESV Program Office, will discuss the "Optimized ESV Specifications."

THE UNITED STATES TECHNICAL PRESENTATION

SECTION TWO

PART 5

ESV TEST PROGRAM

NORMAN S. STAHLER, *Senior Staff Member*
Office of ESV Programs
National Highway Traffic Safety Administration
and
FRED ARNDT, *ESV Program Manager*
Dynamic Science

Mr. Stahler

It is our intent today to bring you up-to-date on what has transpired in the area of ESV testing since the Third International ESV Conference. I would like to again stress the fact that this presentation is a cooperative effort by the Office of ESV Programs and its test contractor, Dynamic Science.

I think it is appropriate to note that Volume I of the *Final Report on AMF Fairchild ESV Testing* has been made available to all attendees. This volume summarizes the test results and is one of four volumes that constitute the full report. The more detailed Volumes II, III and IV are in production and will be available early in April.

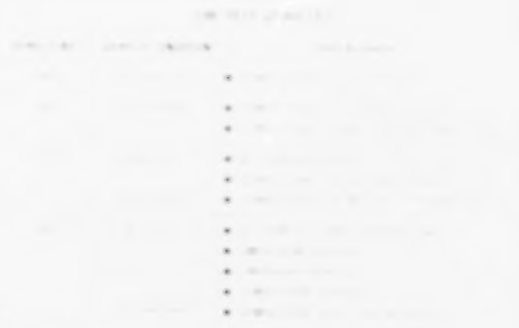

Figure 1

Figure 1 shows the numerical designations of the four GM ESV's that were delivered to the Dynamic Science test facility in July 1972. Numbers 08 and 09 were untested vehicles, while 06 and 07 were previously crash tested by GM during its development program. Both 06 and 07 were suitably and selectively refurbished by GM to allow them to be used in vehicle-to-vehicle crashes. Testing of the GM ESV was completed on schedule by December 5, 1972 (Figure 2).

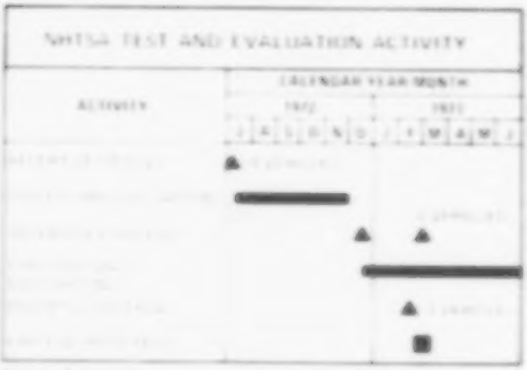

Figure 2

The Ford ESV arrived in Phoenix in December, and is currently undergoing accident-avoidance testing. Brake testing has been completed and the results will be presented later. The Ford Motor Company has graciously offered, and we have accepted, a second ESV, which will be delivered to Phoenix this month for crash testing. We are presently on schedule and anticipate the Ford activity to be completed by June of this year.

On February 14, we had the good fortune of receiving the first of what we hope will be a good number of foreign ESVs, the 1,500 pound class Fiat ESV. Just prior to this conference, we conducted two vehicle-to-vehicle crash tests with the Fiat ESV and the AMF ESV. The first test was a front-to-front crash, at a closure velocity of 73.3 mph. The second test with the same vehicles, was an AMF front-to-Fiat-rear at an impact velocity of 60.3 mph. Since the objective of these tests was to determine structural performance, there were no dummies in either vehicle. Preliminary results will be presented a little later.

Figure 3 summarizes the tests conducted on the GM ESVs as follows:

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EVALUATIONS PERFORMED ON GM ESV'S

WEIGHTS AND MEASURES – ACCIDENT AVOIDANCE

- BRAKING PERFORMANCE
- STEERING PERFORMANCE
- HANDLING PERFORMANCE
- OVERTURN IMMUNITY PERFORMANCE
- ENGINE PERFORMANCE
- VISIBILITY

CRASH INJURY REDUCTION

- 30 MPH FRONT TO SIDE IMPACT
- FRONT AND REAR LOW SPEED IMPACTS
- 50 MPH FRONT BARRIER IMPACT
- 60 MPH FRONT TO REAR IMPACT
- 30 MPH FRONT TO SIDE (AT MILFORD)

Figure 3

Post Crash Factors

The Ford ESV will undergo the same accident avoidance testing as the GM. In the area of crash injury reduction, the Ford ESVs will be subjected to:

1. Front and rear low-speed impacts (10 mph and repeatability)
2. 50 mph front barrier impact
3. 40 mph front barrier impact
4. AMF front to Ford rear at 60 mph

Let us proceed to the actual test results, starting with the weight and measurements. Figure 4 is a tabulation of significant parameters on the GM ESV. The GM ESVs were substantially lighter than the AMF and Fairchild vehicles; however, weight continues to be a problem. The vehicle generally met the dimension requirements with very minor deviations. It had a passenger capacity of five, but the center position of the rear seat would not accommodate more than a 50th percentile male.

The weight and dimensions of the Ford ESV are shown in Figure 5. Generally, the vehicle met all specifications, except for weight. It has the same seating capacity as the GM ESV.

The actual weight of the Fiat, as delivered, was 1,638 pounds. This was a structural vehicle without seats and internal trim.

Mr. Arndt:

We will present a summary of the accident-avoidance results, with emphasis on the braking performance of the vehicles – first, the GM performance results.

WEIGHT AND DIMENSIONS – GM ESV

	REQUIREMENT	ACTUAL
● WIDTH	80 IN	80 IN
● HEIGHT	58 IN	58 IN
● LENGTH	220 IN	[illegible]
● WEIGHT	5000 ± 250 LB	[illegible]
● WHEELBASE	120 IN	120.2 IN
● TREAD WIDTH	64 IN	64 IN [illegible] (REAR)
● PASSENGER CAPACITY	5	5
● CURB LENGTH WHEELBASE	[illegible] MAXIMUM	[illegible]

Figure 4

WEIGHT AND DIMENSIONS – FORD ESV

	REQUIREMENT	ACTUAL
● WIDTH	80 IN	[illegible]
● HEIGHT	58 IN	[illegible]
● LENGTH	220 IN	[illegible]
● WEIGHT	5000 ± 200 LB	[illegible]
● WHEELBASE	120 IN	[illegible]
● TREAD WIDTH	64 IN	[illegible] [illegible] (REAR)
● PASSENGER CAPACITY	5	5
● CURB LENGTH WHEELBASE	[illegible] MAXIMUM	[illegible]

Figure 5

Stopping distance, brake efficiency, and the emergency-braking performance of the GM ESV is shown in Figure 6. It should be noted that, with the exception of the 100-percent weight condition (curb weight plus 1,000-pound load) for the braking-efficiency tests, all of the tests were conducted on a dry asphaltic concrete surface having a nominal skid number of 75.2. For the brake-efficiency tests, the wet surface skid number was nominally 31.9. A 40-percent weight (curb weight plus 400-pound load) condition was used for the remaining tests, with the exception of the minimum-weight, braking-efficiency test which was conducted at curb weight plus 160-pound load.

Stopping distance of the vehicle slightly exceeded the requirement of 155 feet. Braking efficiency, however, was considerably above the 80-percent requirement, at 86 percent.

The vehicle satisfied the specification for brake efficiency for the minimum weight requirement only. Stopping distances for these tests were, as indicated, at 258 feet and 157 feet for maximum and minimum weight conditions, respectively. It should be noted that stopping distance requirements were not specified for these tests.

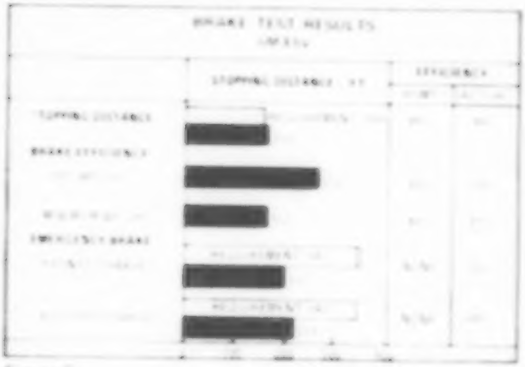

Figure 6

Emergency-brake tests were conducted for two failure mode simulations — fronts brakes disabled and power booster disabled. The vehicle satisfied the stopping distance in both cases with a good margin. The vehicle had a backup, electrically-operated, power-boost pump, which would have taken over automatically had the main booster pump failed. hence, the pedal force did not exceed the 150-pound design requirement. No performance requirement was specified for efficiency; however, we have computed the efficiency and presented it here as a matter of interest.

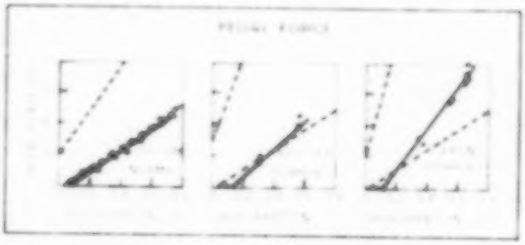

Figure 7

Pedal force gain results are presented here (Figure 7) for the conditions of brake system normal, booster disabled, and front systems disabled. For each case, the specification requirement lies between the dashed lines. The vehicle performed well within the desired 85-pound pedal force goal. Generally, however, the gain was marginally low for the system normal and booster disabled condition.

As far as the remaining GM accident-avoidance performance is concerned, the vehicle generally met the specifications with relatively minor exceptions.

As we indicated earlier, the Ford ESV brake-testing activity has just been completed, and we are only able to present preliminary data at this time.

Stopping distance, brake efficiency, and emergency-brake performance are indicated in Figure 8. Basic-test conditions, as well as performance criteria, are identical to those discussed previously for the GM vehicle. The stopping distance of 157 feet was very close to the requirement of 155 feet. Brake efficiency was 81 percent, exceeding the 80-percent requirement.

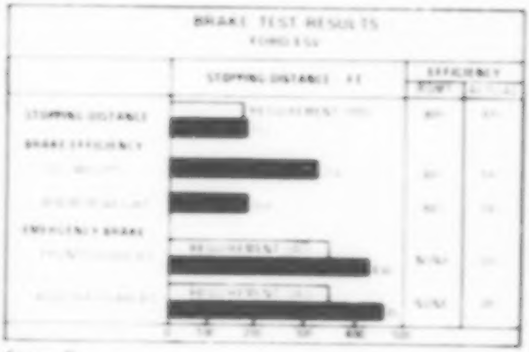

Figure 8

For both the 100-percent weight, wet-surface stops and the minimum-weight, dry-surface stops, the 80-percent efficiency requirement was not met. The minimum weight was 160 pounds, which is equivalent to the 16-percent test weight condition.

For the emergency-brake test conditions with simulated failures of fronts disabled and booster disabled, the vehicle exceeded the 343-foot stopping distance requirement. The pedal force limit of 150 pounds was reached in both cases.

Pedal force gain results are shown in Figure 9. The vehicle met the specification requirements for pedal force normal condition. The nominal pedal force goal of 85 pounds was achieved.

The pedal force front out condition is marginal at the upper limit. Pedal force was not increased above the 150-pound level because the rear anti-skid system was in modulation; therefore, we were close to the maximum braking effort for the vehicle in this condition.

For the condition of no booster, the performance was within specifications to above 0.25 g. Beyond this point, a rapid increase in the pedal force was experienced to the 250-pound pedal-force design requirement. A vehicle deceleration of 0.3 g was achieved at this pedal force level.

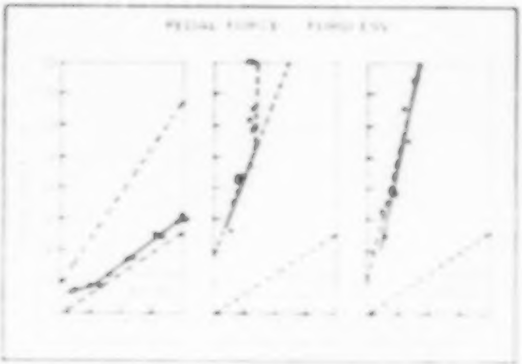

Figure 9

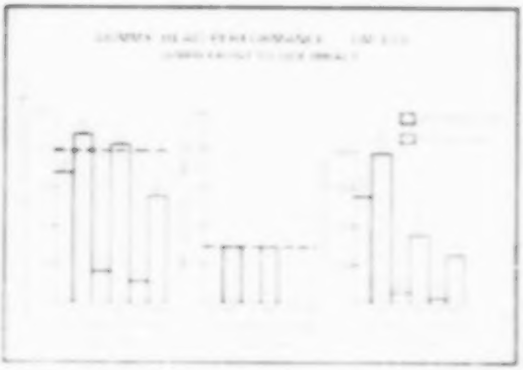

Figure 10

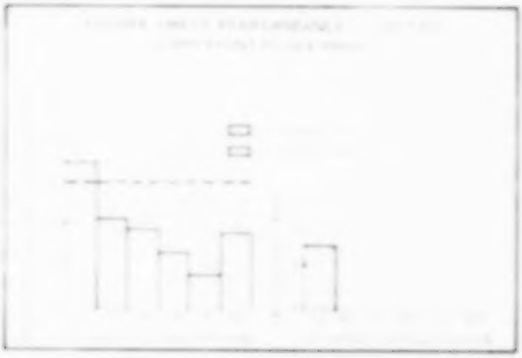

Figure 11

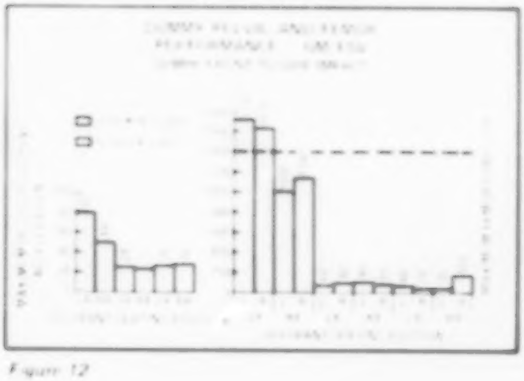

Figure 12

A comment is in order regarding the Ford disabled test results. They [illegible] normal. Subsequent to the testing it was found that a problem existed with the mechanical linkage in the [illegible] pedal design. This caused the abnormally high measured pedal force levels.

Results of the crash injury information tests with the GM ESVs will be covered primarily in the vehicle crash characteristics and for the key occupant performance parameters obtained during the 50-mph barrier test and the 60-mph [illegible] test. Before proceeding with these results, I would like to make a few comments regarding the side impact tests. A 90-degree side impact was conducted at Phoenix with two instrumented GM ESVs (GM 06 and GM 07). This test, at 29.6 mph, was primarily a structural test to evaluate the subcompartment characteristics of the vehicle. Dynamic intrusion was 4 inches as measured at the right front passenger position. Residual intrusion was 2.5 inches at the same location.

The same type test was conducted by GM for the NHTSA at Milford with two instrumented dummies in the striking car and four in the struck car. The impact speed was 29.8 mph, and the structural responses were nearly identical to those obtained from the Dynamic Science Test.

Dummy head performance from both the vehicles is shown in Figure 10. The struck vehicle was impacted on the right side, resulting in high acceleration on the right front dummy's head as it impacted the interior of the vehicle. In the striking car, the right front occupant hit its head against the front dash during impact. With the exception of these two occupants, dummy head performance was within design specification. Chest accelerations were within specification with the exception of the left front occupant in the striking vehicle (Figure 11).

Pelvis and femur performance information is summarized in Figure 12. As with the chest, the performance criteria were met with the exception of the left front dummy in the striking car.

The GM 09 vehicle was subjected to both a low-speed front barrier impact and a low-speed rear barrier impact to evaluate the no-damage requirement of 5 mph set forth in the GM contract. The front impact was performed nominally at 5 mph with a test weight of 5,400 pounds (no dummies), with no damage occurring. The rear impact into the barrier, which replaced the 5 mph rear bogey impact, was conducted at a nominal equivalent impact speed of 3 mph. No damage occurred in this impact.

Let us examine the structural response of the vehicle, as shown in Figure 13. The vehicle exhibited an average deceleration of 26.4 g's with a rate-of-crush onset of 25 g's per foot. Considerable variation was noted in the response as indicated, and the duration of time that the deceleration exceeded 40 g's was 10.5 milliseconds, as compared to the specification requirement of 5 milliseconds.

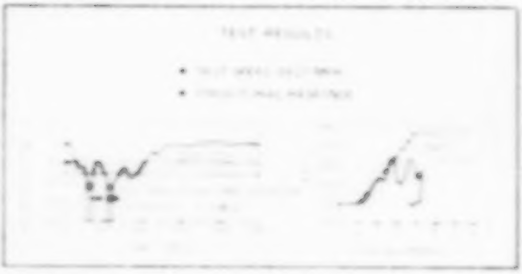

Figure 13

This information reflects average data obtained from accelerometers located in the occupant compartment of the vehicle at the front seat positions.

The vehicle impacted the barrier at 40.27 mph. Five milliseconds following the impact (Figure 14), the bumper-mounted sensors initiated the firing sequence to the inflatable occupant restraint system. Approximately 12-to-13 milliseconds after impact, the air bag system began to deploy, and complete deployment occurred in the 20-to-25 millisecond range. The occupants then began their major interactions with the restraint system. The vehicle was occupied by four 50th percentile anthropomorphic dummies, two located in the front seating positions and two in the rear outboard seating positions. A summary of some of the significant occupant-performance results are now presented.

Figure 14

These data (Figure 15) represent head performance with respect to peak acceleration and accumulative time over 80 g's. We also show the severity index which was not cited as a performance criterion in the development contract. With the exception of the right front occupant, all performance criteria were exceeded. Generally, the dynamic response of the head of the occupant had two major pulses. One was associated with the initial deceleration of the occupant, and the second was associated with rebound as the occupant was tossed back into the seat, resulting in the head impacting the headrest. A significant portion of the accumulative time over 80 g's is associated with the rebound, as shown by the cross-hatched area in Figure 15. Peak resultant accelerations were also very high during the primary pulse as well as during the rebound; the dashed lines show the rebound peaks. It is interesting to note that values for the right front occupant are generally lower than those for the remaining occupants. This improved performance can probably be attributed to a tearing of the right front air bag, which occurred during the crash event at approximately 60 milliseconds following impact, and resulted in a generally softer bag for the dummy. The dummies did not bottom out against the front portions of the car during the crash.

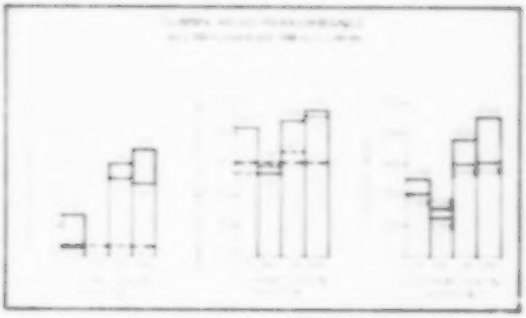

Figure 15

Figure 16 shows chest-acceleration performance for each of the occupants. The peak resultant accelerations were exceeded in all cases. No accumulative time is specified, however.

The pelvic-and-femur load performance are shown in Figure 17. No performance criteria were specified for the pelvic acceleration, however, there is a correlation between the femur load levels and the pelvic acceleration levels. Femur loads were generally within compliance for the rear occupants and were exceeded for the front occupants. The pair of bars for each seating position represent right and left femurs, respectively. High-tensile loads were noted on the left front occupant where the knees of the dummy actually pushed through the energy-absorbing knee panel and became wedged. We also noted an asymmetrical loading effect on the rear dummies where both the right femur of the right rear occupant and the left femur of the left rear occupant were generally higher than their respective inner femurs. This was correlated with relatively high-lateral accelerations in the pelvic regions and is attributable to what we believe is a so-called "wedging effect" of the single rear air bag. We believe that this would not have occurred if a third dummy had been positioned in the center rear seating position. This conclusion is also supported by data which we derived from the front-to-rear test.

The speed of the impacting test vehicle was 63.8 mph. The striking vehicle (GM 08) exhibited a static crush of 30 inches, while the struck vehicle (GM 09) experienced 46 inches of crush. Occupant compartment accelerations on both vehicles were nominally at 12 g's.

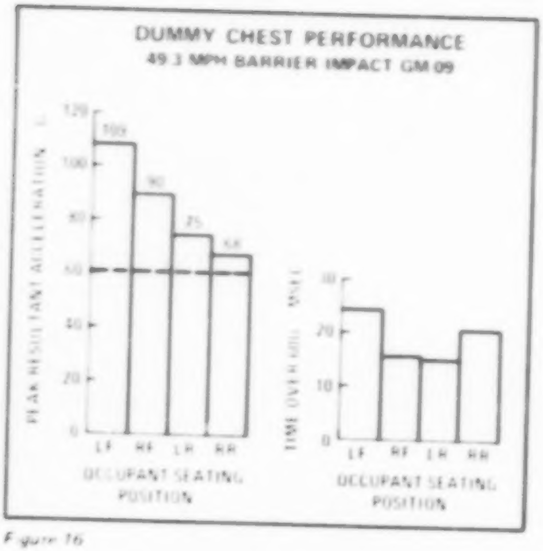

Figure 16

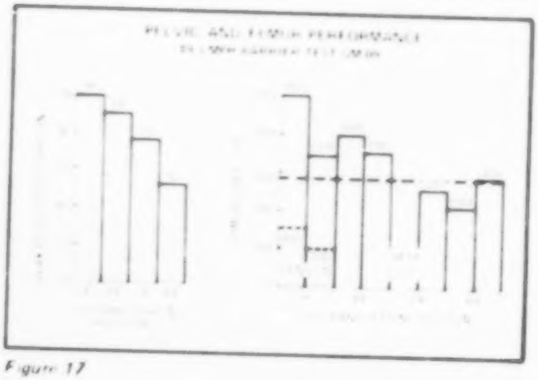

Figure 17

The functioning of the inflatable restraint system was very similar to that of the 50 mph barrier test, as evidenced by the sequence of events being within a millisecond or two of the barrier impact.

Dummy-head data generally met the design requirements (Figure 18). A relatively minor exception occurred with the left rear occupant, in that both design criteria were exceeded during the rebound.

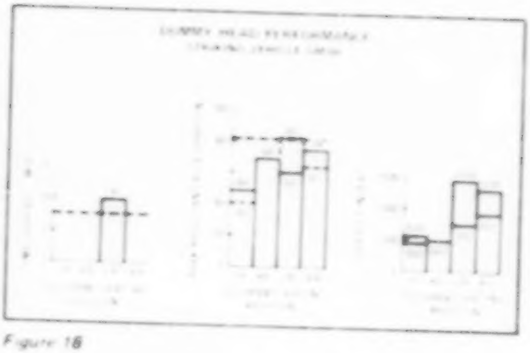

Figure 18

As with the 50 mph barrier test, the right front air bag experienced a failure. It was a seam failure rather than a tear that occurred during the 50 mph barrier test. A significant difference between these two tests is that the right front position pressure-relief ports did not blow out in this test, but they did in the barrier impact. The right front occupant generally fared better than the others, since no rebound occurred.

Dummy-chest performance, as shown in Figure 19, met the design requirements satisfactorily in all cases.

Pelvic-and-femur performance are shown in Figure 20, and, as with the barrier test, the pelvic and femur loads generally show the same correlating relationship. Left front femurs were high and exceeded the specification. The outer femurs of the respective rear seat dummies were marginal, as far as the specification was concerned, and were again asymmetric.

Two dummies were in the rear of the struck car (Figure 21). Each contained accelerometers in the head cavities only. Data derived from these dummies are high and rebound accelerations of significance occurred as the occupants hit the rear of the front seat.

In the post-crash factors area, the vehicle performed well. The fuel system was satisfactory, and

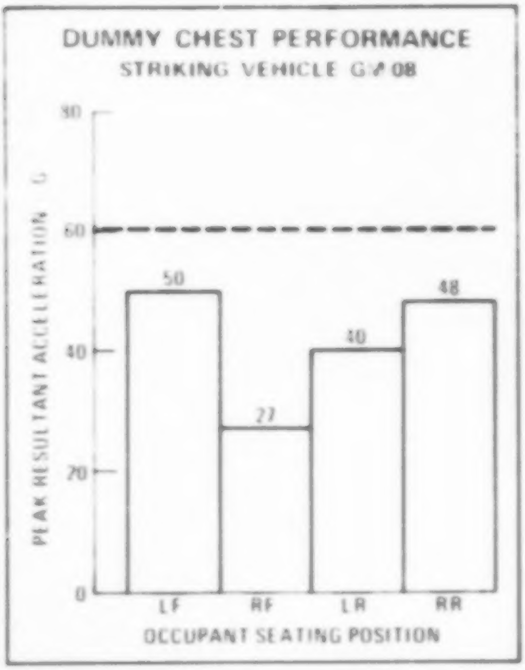

Figure 19

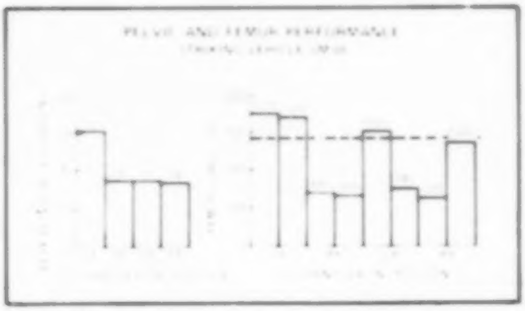

Figure 20

simulations of emergency rescue of occupants occurred without any major difficulties.

Other data of interest are presented in Figures 22 and 23. The dynamic-pressure rise within the occupant compartment measured at the front and rear seating positions is shown in Figure 22; we note a peak DC pressure rise of 1.75 psi at both seating positions. These particular data, which were derived from the 50 mph barrier test, are identical to the data obtained from the 60 mph front-to-rear test. Unfortunately, mechanical failure occurred in both microphones, and we were not able to obtain what we believe to be reliable sound measurements from the 50 mph barrier test. However, we did successfully obtain sound data from the 60 mph front-to-rear impact which is shown in Figure 23. Peak sound levels were measured at 165.2 DB at the front seating position and 161.4 DB at the rear seating position. Since the pressure responses measured from both the barrier test and the rear-end vehicle test were nearly identical, we believe that it is reasonable to assume that the DB levels were also close.

As we indicated earlier, two crash tests were conducted with the Fiat. A summary of some of the test results is shown in Figure 24.

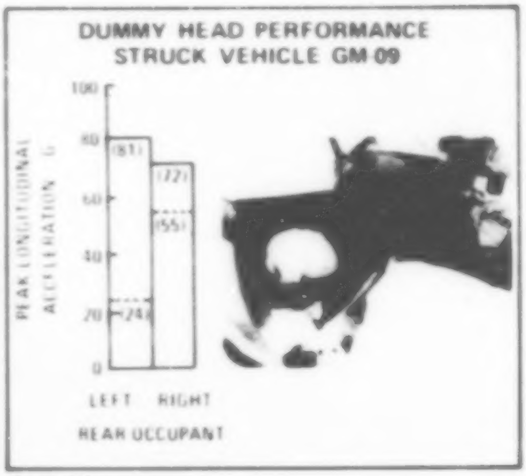

Figure 21

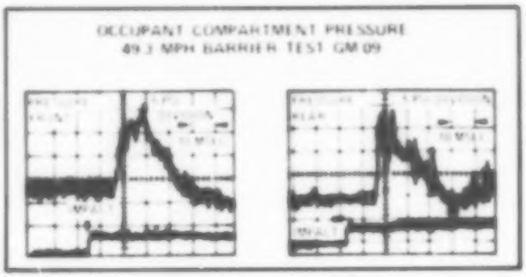

Figure 22

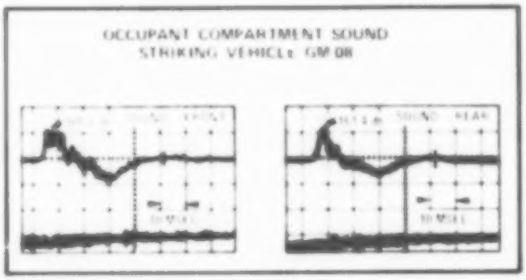

Figure 23

TEST RESULT SUMMARY
FIAT AMF ESV

TEST CONFIGURATION	FRONT TO FRONT	FRONT TO REAR FIAT STATIONARY
[illegible]	[illegible] MPH [illegible]	[illegible] MPH
[illegible]		
[illegible]	[illegible] IN	[illegible] IN [illegible]
AMF	[illegible] IN	[illegible] IN
[illegible]		
[illegible]	[illegible]	[illegible]

Figure 24

The vehicle weight ratio for this test was three-to-one (3:1) with the AMF ESV weighing 5,266 pounds and the Fiat ESV 1,753 pounds. Closure velocity of the two vehicles was 73.1 mph. The total-system dynamic crush for the front-to-front test was 54.2 inches with a split as indicated. This was very close to the crush we predicted. The dynamic crush of the Fiat was 18 inches and the post-test static crush was approximately 10 inches.

As yet, we do not have dynamic crush information from the 60.7 mph front-to-rear crash; however, the static crush on the Fiat was 10 inches. Intrusion into the occupant compartment can be considered minimal. The seat intrusion was a result of localized impact from the rear-mounted engine. The front intrusion was fairly uniform, indicating a general foreshortening of the occupant compartment as a whole.

Figure 25 indicates the dynamic response of the vehicles during the front-to-front impact. The crushing force during the test was nominally at 90 KIPS. The AMF hydraulic-system force matched the Fiat crush characteristics as anticipated. The velocity change was 60 mph for the Fiat with a velocity change of 20 mph for the AMF, indicating that the collision was nearly plastic.

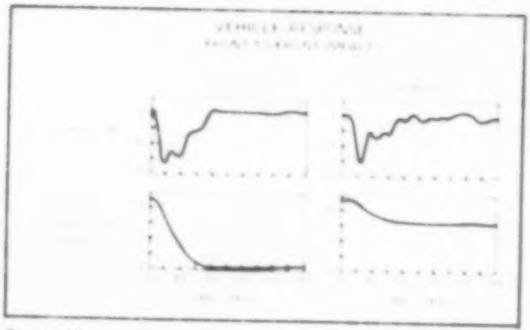

Figure 25

The Fiat ESV experienced an average acceleration of 34.1 g's with the AMF ESV decelerating at an average of 10.6 g's.

Similar data are not yet available for the front-to-rear test.

Mr. Stahler

I believe that there is general agreement on the extreme difficulty of correlating barrier-crash data with car-to-car crash data, especially when the variety of vehicles and possible crash configurations with those vehicles are considered. Related to this is our concern over front-end aggressiveness which dates back to 1968. What can be done to assist in solving these problems?

Under the auspices of other NHTSA programs, a two-pronged effort is underway: (1) to better define the problem in an analytical sense, and (2) to check the results of the analyses by means of improved test methodology. The analyses are continuing, and, in the meantime, new test devices have been developed by Dynamic Science. One device is the variable-rigidity barrier and the other is a variable-rigidity bogey vehicle. With these devices, the bogey bumper and barrier face yield in a programmed manner during impact. Two basic systems, fixed force and hydraulic, can be accommodated on both test devices.

The fixed-force system uses crushable material, such as aluminum honeycomb, to obtain the desired force deflection characteristics. The types of response which can be programmed into the system are shown in Figure 26. We see a family of force responses with

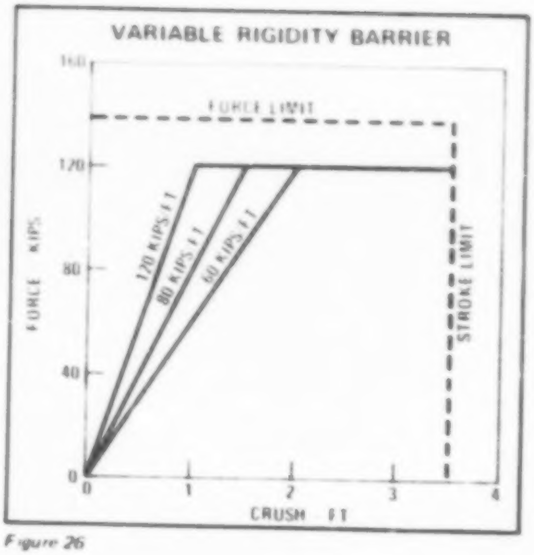

Figure 26

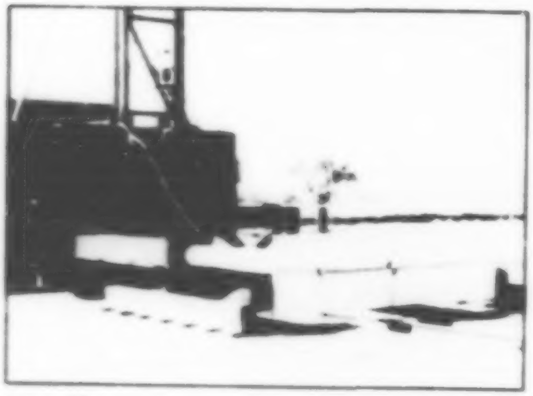

Figure 27

[illegible] of 120 KIPS, 90 KIPS, and 60 KIPS per [illegible] which [illegible] at [illegible] of 120 KIPS. Figure 27 shows the [illegible] on the barrier.

The [illegible] vehicle equipped with the [illegible] [illegible] AMF prototype ESV is shown in Figure 28. [illegible] was given consideration [illegible] which provide the flexibility of [illegible] [illegible] and [illegible] [illegible] characteristics to evaluate their effect [illegible].

Figure 28

By no means are we claiming a panacea for statistical methodology, but we do feel that definite progress has been made. We also acknowledge that there is much more to be accomplished in improving accident avoidance test techniques. I assure you that we intend to continue our efforts in both areas and, we encourage others who are industriously working on this problem to continue their efforts and to report their findings.

With the advent of testing our first non-American ESV, the Fiat, I would like to conclude with an invitation to the other participants in this program to send us their ESVs for testing and evaluation. Cooperation of this kind will definitely enhance the mutual understanding and communications that are so vital to the success of our endeavors.

THE UNITED STATES TECHNICAL PRESENTATION

SECTION **TWO** PART **5**

CONCLUSIONS AND PROJECTIONS ON ESV DEVELOPMENT

LOUIS C. LUNDSTROM, *Director*
Automotive Engineering Safety
General Motors Corporation

General Motors completed the development and testing of its Experimental Safety Vehicle prototypes in May 1972. Final reports, recommendations and other program documentation required under our contract with the United States Department of Transportation were delivered on schedule. On June 30, 1972 two ESV prototypes, and two ESVs rebuilt after testing by General Motors, were delivered to the U.S. National Highway Traffic Safety Administration for its independent test and evaluation. Arrangements also were made for the loan of a fifth ESV for demonstration (Figure 1).

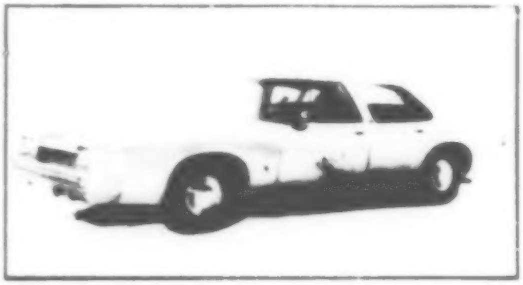

Figure 1

The GM-ESV has been described in detail in previous reports, as has been the test data relating to accident-avoidance capability, vision, structural integrity, occupant crash protection and other requirements of the program. In general terms, the GM-ESV designers produced a vehicle that was very close to the contract specifications for accident avoidance, and was close to the specifications for occupant crash protection, but as a result was seriously overweight. The purpose of this report is to comment on the results of the first ESV program, to add comments relative to the recently published specifications and goals for a 3,000 pound intermediate ESV, and to suggest areas for additional research in highway safety.

The very detailed specifications for the first ESV program have been well publicized. The contractors attempted to meet each one of them. However, occupant crash protection was to receive major emphasis. This was done. Early in the design phase, it became evident that it would be extremely difficult, if not impossible, to meet all of the specifications. The designers in each company with ESV contracts then placed top priority on occupant crash protection and exterior dimensions, and permitted vehicle weight to increase beyond specifications. At that time, it was the only reasonable method of continuing the design, build and test program. We must be careful, however, to point out that it was not possible to obtain the specified occupant crash protection and structural requirements at a weight that would be considered practicable for today's vehicles or for mass-produced vehicles of the future.

The GM-ESV, after a redesign and incorporation of aluminum and high-strength steels, weighed 5,050 pounds. This was considerably lighter than the experimental safety vehicles built by other companies with more conventional materials.

The first GM-ESVs provided an opportunity for our designers to apply some recently developed analytical modeling related to vehicle safety and to demonstrate the tradeoffs between the requirements for passenger comfort and convenience, and the requirements for accident avoidance and occupant crash protection.

We have an extensive film library documenting the 12 complete vehicle impact tests that were conducted at the GM Safety Research and Development Laboratory before prototype delivery to the U.S. National Highway Traffic Safety Administration. Some of the results of those tests were:

50 mph Front Barrier Impact This test took place after our change from a 10 mph to a 5 mph bumper system, which proved worthwhile in controlling the vehicle's pitching motion.

50 mph Front Pole Impact In this test, hitting the pole at 50 mph resulted in 52 inches of deformation. The interior crush of 11.8 inches was limited to the center of the vehicle, and was reduced to

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less than three inches of intrusion at the front occupants' [illegible] foot locations.

[illegible] mph Car-to-Car Impact — Front-to-Rear — In this test, crush was 30 inches in the front of the striking ESV and about 40 inches into the rear of the struck ESV due in part to bumper override even though the bumpers were designed with identical face widths. The air cushions deployed properly in the striking car and occupant performance was [illegible] specification for the [illegible] and [illegible] pounds in the striking car.

[illegible] Side Impact — This test consisted of a car striking the side of [illegible]. The [illegible] [illegible] away from [illegible] on the [illegible]. The [illegible] were generally [illegible] proper [illegible] the [illegible] [illegible] [illegible] [illegible] [illegible].

[illegible] Car-to-Car [illegible] — [illegible] [illegible] [illegible] [illegible] design [illegible] [illegible] the [illegible] [illegible] [illegible] [illegible] the [illegible] and [illegible] [illegible] speed [illegible] slightly [illegible] of the struck car [illegible] were kept in place by the [illegible].

It is important that we do not consider that the data submitted [illegible] would function effectively in highway vehicles. Although General Motors is interested in developing passive restraint systems like the air cushion, and it has 1,000 cars with front seat air cushion restraint systems in service today, we are not confident that the technical capability exists to build such systems into vehicles to protect occupants in the very severe 50 mph barrier equivalent range required of the ESV program. The ESV-1 program, for example, did not consider high-speed air cushion inflation hazards to out-of-position occupants or small children. The contract test specifications called only for the range from a 5th percentile female to a 95th percentile male. Furthermore, the possibility that an air cushion, designed for a 50 mph barrier impact, may be required also to inflate too violently in lower speed impacts, was not considered.

Data submitted to NHTSA through our comments to the several dockets on Standard 208, expressed our concern for inflation hazards, as well as for the control of noise at inflation rates, even for 30 mph protection. Following many tests not required of an ESV program, we believe that we have a front seat system ready for public use with a capability for 30 mph barrier equivalent protection. Although we have inflated the air cushions in our ESV cars in the much more severe 50 mph barrier equivalent accidents, we seriously question the [illegible] high-speed inflation hazards [illegible] and expensive noise.

The original ESV contract specifications were deficient by not requiring a consideration of such potential problems. [illegible] DOT specifications for the ESV did not include a request for measurement of noise levels resulting from deployment of the air cushions, nor were any test procedures established by the contract. Likewise, the NHTSA did request that [illegible] be made, but did not specify measurement procedures.

Limited test data obtained by those of the ESV contractors during their development work indicated that the difficulty is a real possibility, and if so there is a need to correct test equipment and test procedures. If we are [illegible] to protect [illegible] [illegible]. Air cushion systems for 50 mph barrier impact protection require further development to reduce the effect of the noise [illegible], the deployment hazard and [illegible] noise.

In previous ESV conferences, [illegible] sensors were discussed as a means of compensating for rapid inflation, and General Motors, as well as other companies, is continuing to investigate this possibility. Thus far, we have been unable to find a sensor that can predict the forces with which an obstacle might be hit so that air cushion inflation speed can be modulated accordingly. Until such problems are solved, the air cushion restraint system does have practical limitations at higher impact speeds.

The new intermediate ESV specification could be improved by requiring specifications for testing noise and overpressure, as well as any inflation hazard that might relate to both out-of-position occupants and children. Without such specifications, the ultimate, real world usefulness of the results of still another ESV program will be limited.

Another requirement neglected in the intermediate ESV specification — but obvious in a car designed for production — is the consideration for occupants of grossly different weights and dimensions. All testing is required with one dummy size — the 50th percentile male. As a result, no data will be available to indicate the response of occupants of other weights. It is worth noting, in fact, that no data yet have been found to indicate whether established human tolerances, as defined in terms of accelerations and forces, are independent of the sizes of the occupants. Also, the intermediate ESV requirements call for a maximum rate of onset of restraint forces of 1,500 g/sec., and for a maximum interior surface force of 50 psi. These are at least controversial and in need of considerable research. Such data could have a tremendous influence on the design of restraint systems.

If you will recall, I said that occupant crash protection features for the original ESV-1 were given priority over weight. It now appears that the reverse is true for the intermediate, 3,000 pound ESV, and weight becomes a top-priority item. Our experience with the ESV-1 leads us to believe that a designer will find it virtually impossible to meet all of the specifications that have been proposed for the intermediate, 3,000 pound ESV. As a next step, then, serious consideration should be given to an identification of priority items, such as weight and exterior dimensions, while allowing the designer the option of providing as much occupant crash protection as the design parameters permit, even if it be something less than 50 mph barrier impact. The intermediate ESV program will not likely receive serious consideration until a compromise of this type is permitted.

Now let me discuss some of the more innovative features of the GM-ESV and what we believe is their potential.

The GM-ESV mirror system, and the provisions for controlling the size of the eye ellipse, provided significantly improved rear vision, without resorting to the more complicated and higher cost periscope system. Of particular significance was the smooth interior and breakaway mirror for added occupant protection in rollover accidents (Figure 2).

The four-wheel, anti-lock brakes with backup power assist demonstrated features that extend current brake technology to its limit (Figure 3). Radar-triggered braking was not considered because of the uncertain benefits and problems. Responses to the NHTSA docket on automatic-braking systems provide current information on this subject. It is now important to determine the benefit of such systems in highway use and establish their benefit/cost relationship with some degree of certainty.

The elimination of the front pillar on the GM-ESV would have necessitated a considerable increase in the weight of the roof structure, if steel had been used instead of aluminum (Figure 4). Tradeoff studies should be made to determine the effectiveness of this design in both accident avoidance and occupant crash protection.

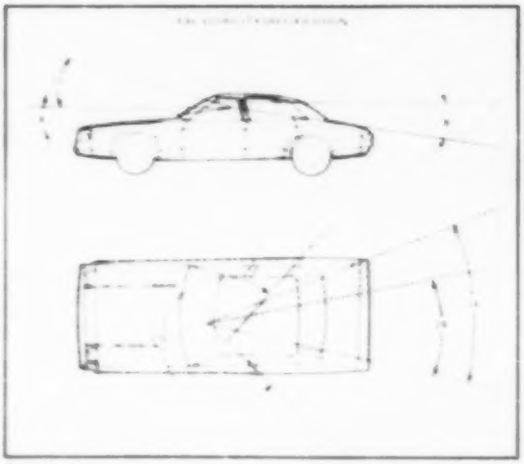


Figure 2

The message center concept demonstrated on the GM-ESV requires additional research to determine its value for the safety of the motorist (Figure 5). Such distribution of primary and secondary information is convenient, but its need as an assist in accident avoidance has not been established. The SAE Vehicle Research Institute has initiated a research project relating to this important area.

Bumper requirements for both safety and the reduction of vehicle damage have been discussed in previous ESV reports (Figure 6). We believe the continued development of bumper specifications, as proposed in the intermediate ESV program, are desirable, but limited, in value. Because many new bumper concepts are under consideration at this time, the intermediate ESV program should leave this specification open. This would enable the designer to use the latest possible developments in providing a bumper most desirable from a protection and bene-

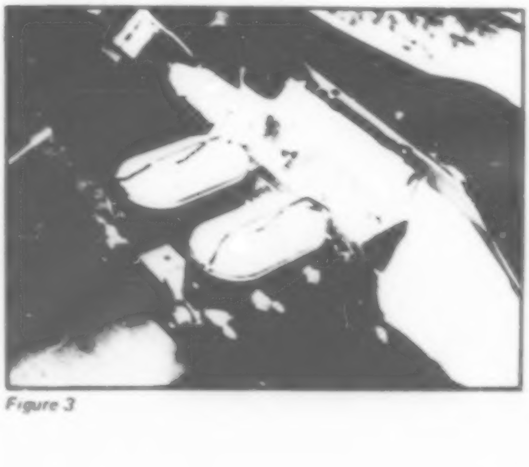


Figure 3

fit cost standpoint, rather than requiring him to simply meet what appears to be a desirable specification at this time.

The high-level rear signaling system demonstrated on the GM-ESV-1 is somewhat similar to the lighting proposed by the Safety Administration as an option on 1974 vehicles and a requirement on 1977 vehicles (Figure 7). Although designers probably could provide such high-level signals at some additional cost and complication, there has been too little data accumulated to determine their effectiveness in reducing accidents. The designer will have no difficulty providing this type of system on the 3,000 pound ESV; however, we would encourage NHTSA to conduct parallel studies of their effectiveness and benefit cost. As a matter of fact, new ESV programs should be applying lighting and signaling systems well beyond the current state-of-the-art, rather than demonstrating an ability to comply with a published proposed safety standard.

One of the major concerns in our development of the GM-ESV-1 was the type of interior features needed to provide the required occupant crash protection (Figure 8). Because of the shortness of contract time, the designer was forced to give occupant crash protection top priority, and had to devise a necessary combination of padding, energy absorption and air cushions to meet this requirement. From a technical standpoint, it can be said that he was reasonably successful in meeting the specifications. However, the specifications were deficient in not recognizing the minimum space that the U.S. car buyer demands. The very large majority of people viewing the car, when it was displayed at the Third International ESV Conference and many auto shows, commented that they would not consider purchasing such a vehicle, largely because of its restricted space and inconvenience of entry and exit. For these reasons and others, this particular ESV design would have to be rated as unacceptable for future production vehicles. The designers who eventually will be developing the 3,000 pound ESV should be required to provide the minimum acceptable interior space necessary to market such a vehicle. The specifications themselves could be specific in this regard, or the designer could be instructed to provide what he believes is the most satisfactory interior design. We fully recognize that maximum occupant crash protection can be provided by restricting space around the individual. This was demonstrated in the GM-ESV-1. But, as noted, this is not a marketable solution, and the next ESV program should demonstrate the degree of occupant protection that can be obtained with a reasonable interior environment.

Figure 4

On still another point, the U.S. ESV-1 program did not permit an evaluation of *active* restraint systems. This, unfortunately, made it too design restrictive. We would propose that the intermediate 3,000 pound vehicle specifications provide consideration of both passive and active restraints. The reason for this recommendation is that one of the major objections that the GM-ESV-1 received from the public was its very high seat backs. These were required to provide reaction members for the rear seat, passive protection air cushions. Until some alternate passive restraint system is developed, which doesn't require such a structure, it may be that the only way to provide the customer with a comfortable rear seat environment is to permit *active* belt systems. Additional remarks relative to the passive restraints and active belt

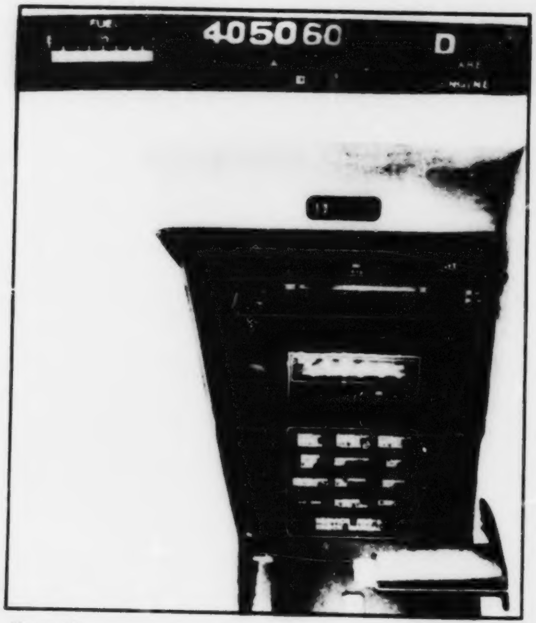


Figure 5

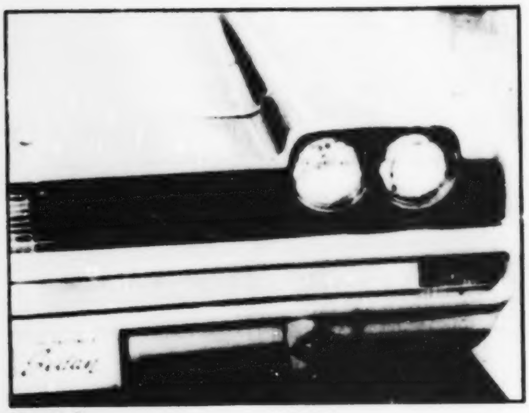


Figure 6

benefits will be given by Mr. Ray Fischer at the Occupant Protection Seminar on Friday of this week.

Actually, a better understanding of restraint system effectiveness is necessary before specifications for future safety vehicles can ever be finalized. Such an effectiveness study requires a national accident data system that can effectively organize the many data sources now in existence and make their results more useful. We have suggested a multi-level program that would provide (1) exposure data, (2) level-one accident data, (3) level-two accident data, (4) special accident studies. With an improved data base, accident causation studies and system effectiveness studies can be determined with greater accuracy than has been possible in the past. For example, with regard to the specifications goals of the intermediate, 3,000 pound ESV program, numerous references are made to the requirement for benefit/cost analyses in the selection of final designs. It is difficult to comprehend how accurate studies can be made unless steps are taken now to obtain additional accident data. A good data base may provide the assurance that is needed to adequately assign safety priorities to the many possible tradeoff evaluations.

General Motors is collecting as much data as it can, relative to its own products, and is conducting initial causation studies that should relate to future vehicle safety programs. The results, hopefully, will provide the improved data base that will enable us to place greater confidence on benefit/cost studies. An example of this will be discussed by Mr. John Rosenkrands during the Accident Avoidance Seminar on Friday.

In regard to handling, it is interesting to note that the specifications for the 3,000 pound ESV do not recognize the possibility for further improvements in tires. Only current production tires are permitted.

Figure 7

Variations in testing have presented another difficulty. A review of the impact tests of the first experimental safety vehicles indicates that there is considerable variation in the results obtained at different laboratories. Part of this can be explained because of the use of entirely different dummies. In addition, we do not know how repetitive tests at the higher impact levels actually can be. In the ESV program, a very limited number of test samples were available. Only in the GM program were duplicate tests run both at the Milford Proving Ground and at the Dynamic Science facility in Arizona.

Still, variations in results from laboratory to laboratory are of less concern than our inability to predict human injury, based upon the test dummies now in use. At the present time, we must recognize that any test development, whether engineered by an auto manufacturer for a normal vehicle or engineered for an experimental safety vehicle project by a non-automotive company, will suffer in terms of its

Figure 8

potential safety until more accurate test tools and simulators are available. For this reason, General Motors has engaged upon a very active program, first to determine the tolerance of humans to impact forces, and second, to improve the simulation of humans through the use of improved test equipment. Future ESV program can presumably be updated as new knowledge is gained and new equipment provided.

One final comment on the proposed intermediate experimental safety vehicle. NHTSA is making a good initial step in inviting comments prior to the next phase of its program. These comments undoubtedly will be varied and, at times, controversial. A formal GM position relative to the overall project has not yet been formulated. I am anticipating that we will recommend that a number of component and background studies be made before letting contracts for another demonstration ESV.

The auto industry has made marked progress through component evolution, rather than complete vehicle redesign. This leads me to believe that the construction of still another ESV, even though of a different size, is not likely to achieve improvements in highway safety commensurate with the time and costs involved. A far greater influence on near-term vehicle safety could result from tradeoff studies, as well as new component and systems development, rather than what could be achieved by building another demonstration vehicle. As new safety systems and vehicle safety components are developed and proven, they then could be added into regular production at times made appropriate with forward planning. Another demonstration ESV might be more justified at some future time when the specifications for such a safety vehicle are supported by research findings and practical considerations.

SECTION TWO

PART 5

RESULTS OF THE FORD MOTOR COMPANY ESV DEVELOPMENT

J. D. COLLINS, *Chief Research Engineer*
and
K. H. ARNING, *Manager*
Special Vehicles Engineering Department
Ford Motor Company

Ford Motor Company entered into a contract with the U.S. Department of Transportation in July of 1971, to develop, build and deliver one ESV prototype within a period of 18 months. We fulfilled this obligation, through the efforts of many people and organizations within Ford Motor Company, with the delivery of one fully-styled and functional ESV prototype to the DOT on December 19, 1972. At this time, the Ford ESV is undergoing testing at the Dynamic Science Test Facilities in Phoenix, Arizona. These tests will culminate with a 50 mph barrier crash scheduled for May of this year.

Our presentation today will include a brief review of the final design configuration of the Ford ESV, followed by a report of the results of significant crash tests and performance tests in the "crash avoidance" category. We will conclude the presentation with a summary of experience gained from our participation in the ESV program and finally will express our thoughts and concerns about the future of the ESV program.

As we have previously reported, our basic objective of the Ford ESV program was to explore the ramifications of meeting or approaching the DOT specifications with designs capable of production in quantity by proven manufacturing techniques and materials. Accordingly, we chose a production sedan design as our point of departure and, in describing our final ESV design, we will compare it to the design of a current Ford sedan.

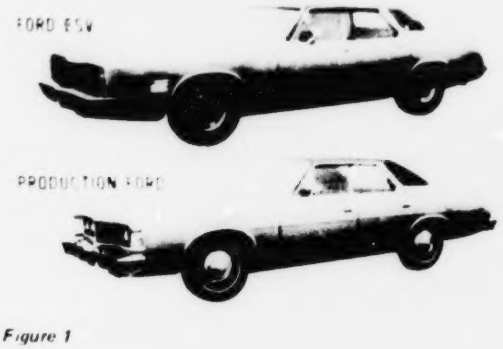


Figure 1

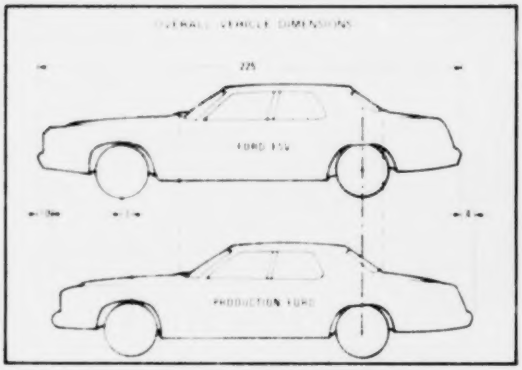


Figure 2

In Figure 1, the ESV is above and a production Ford is below. There is an obvious family resemblance between the two vehicles, but the front and rear length proportions are markedly different.

The front end of the ESV was lengthened 10 inches over the production design, and the wheelbase was increased three inches (Figure 2). This additional crush distance was provided to accommodate a frontal impact at 50 mph. To limit the extent to which overall vehicle length exceeded DOT's specifications, the rear overhang was shortened four inches.

Although the additional 10 inches of front end length would imply a roomier engine compartment (Figure 3), that is not the case, for two reasons. First, the 10 mph "no-damage" bumper system incorporated in the front end requires a bumper movement of approximately seven inches. Second, the addition of the corrugated, controlled-collapse apron structure in the body front end – a feature of the Ford ESV which has been described in previous presentations – encroaches on engine compartment space. The

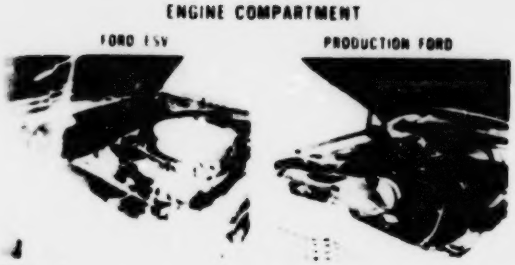


Figure 3

radiator, headlamps, and much front end sheet metal are positioned in essentially their conventional locations.

Shortening of the rear of the ESV body by four inches, and the necessity of recessing the rear lights to accommodate the seven inches of rear bumper movement — again for the 10 mph "no-damage" bumper specification — limits trunk space in the ESV, when compared with our conventional production vehicle (Figure 4).

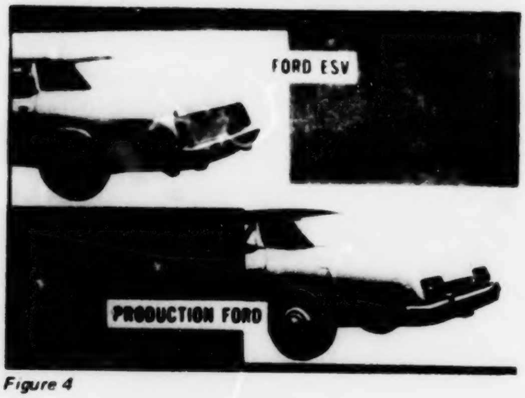


Figure 4

Figure 5 illustrates one of the drastic consequences of the reduction in effective rear length in the ESV. More than 60% of the trunk space has been lost. An inflatable spare tire is used to save trunk room. A regular tire could not have been fitted under the trunk floor. The production Ford has the spare tire located flat over the rear axle, whereas this space is occupied by the fuel tank in the ESV.

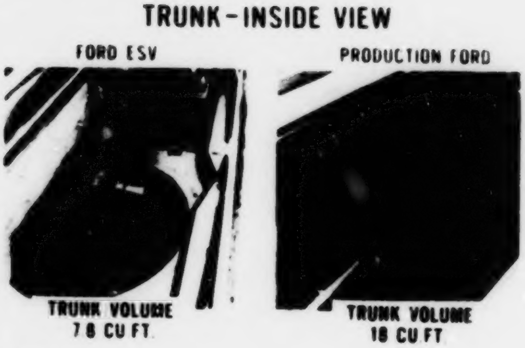


Figure 5

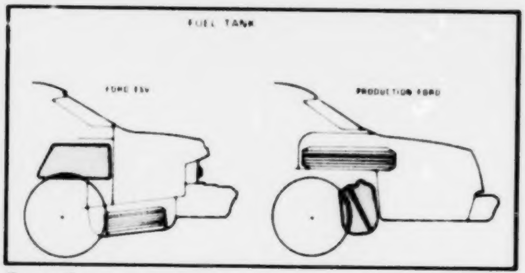


Figure 6

The fuel tank has been relocated from behind the rear axle to a position over the axle to assure compliance with the 50 mph rear moving barrier impact requirement (Figure 6). The construction of the Ford ESV fuel tank is similar to that of fuel cells manufactured for modern race cars. It consists of a heavy-walled, plastic outer shell over porous-foamed plastic on the inside. The fuel tank capacity is larger than that provided in the production Ford, to compensate for the increased rate of fuel consumption due to the increased weight of the ESV. As designed, the experimental vehicle meets the requirement of a 250-mile driving range at 60 mph cruising speed — a specification of our contract with DOT. The reduction of usable trunk volume is again highlighted in this illustration.

INTERIOR PACKAGE

Occupant space in the Ford ESV is restricted, as compared to that of the production vehicle, by the padding and structure employed to help the passenger compartment withstand the severe high-speed test impacts specified in the contract.

Although the car is larger and heavier than a conventional Ford four-door sedan, its maximum capacity is five passengers.

We adopted the "fixed front seat" concept for our ESV design (Figure 7). This permitted a transverse structure through the front seat back, between the "B" pillars, to strengthen the body shell and support the large air bag for the rear passenger compartment structure. Provision for this transverse structure necessitated an increase in seat back thickness of approximately 1.5 inches. We took this 1.5 inches out of the front passenger compartment in order to preserve an acceptable minimum level of leg room in the rear passenger compartment. Consequently, the

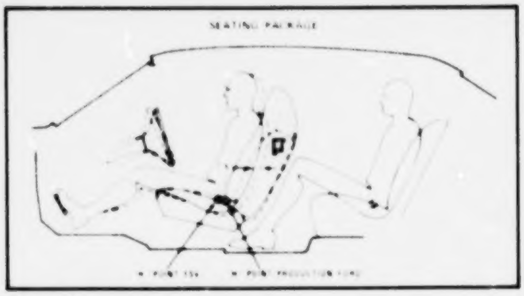


Figure 7

driver and front passenger positions in the ESV are 1.5 inches forward of the extreme rear travel position permitted in the production vehicle.

The fixed-seat design required that movable foot controls be adopted to accommodate a wide range of drivers. The steering wheel, although oriented more nearly in a vertical plane, is positioned relative to the driver very much as in a production Ford car. A vertical seat cushion adjustment of two inches is provided for the driver's seat to optimize visibility for the smaller driver.

This view into the driver compartment of the ESV shows a prominent center console (Figure 8). This feature is actually an important structural member in the vehicle. The size of the console, plus the thick padding applied to the inner door panels, tends to make the front compartment seem more confining that it actually is.

Seating space for the front passenger is the same as that for the driver (Figure 9). The large panel in front of the passenger conceals an air bag and pressure vessel for the restraint system. The traditional glove compartment had to be sacrificed.

As indicated earlier, leg room in the rear passenger compartment is comparable to that of a production Ford with the front seat adjusted to the rearmost position (Figure 10). Thus, the impression of being confined may be largely an illusion, resulting from the unusually high front seat back. Side padding does decrease the seat width appreciably, however, and this compromises seating comfort when three adults occupy the rear seat.

The roll bar in the roof and the high front seat back both reduce forward vision for rear seat passengers (Figure 11). The roll bar extends to the reinforced B-pillars. Figure 12 gives an indication of

DRIVER COMPARTMENT

FORD ESV
PRODUCTION FORD

Figure 8

FRONT PASSENGER COMPARTMENT

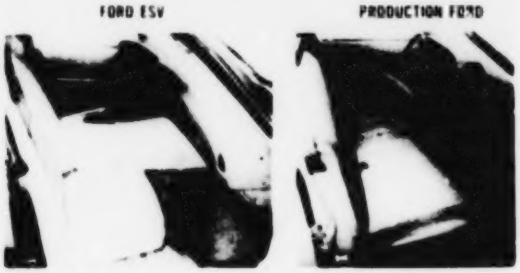


Figure 9

REAR COMPARTMENT

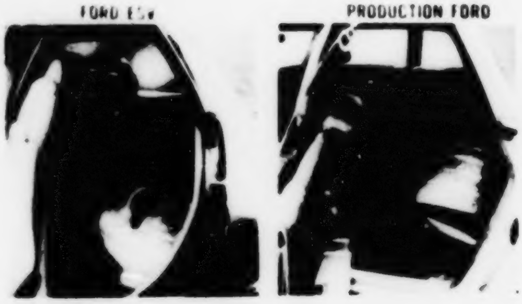


Figure 10

Figure 11

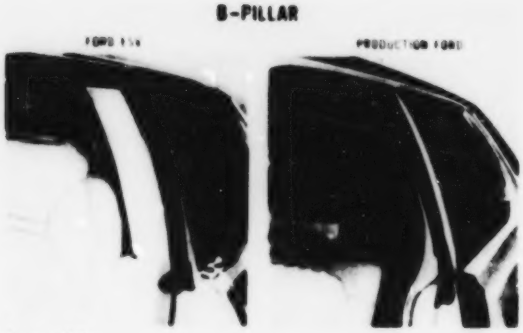


Figure 12

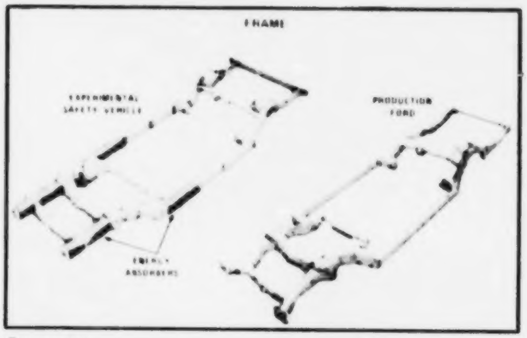


Figure 14

the degree to which forward visibility has been reduced for the rear occupants.

The instrument panel area (Figure 13) presented the most difficult task in packaging. In this normally crowded section of the car, space had to be found for the addition of two air bag systems, without compromising all the other functions normally provided for in this area. One large bag system was fitted behind the right side of the panel for the front passenger. A smaller bag system was fitted on top of the left side over the instrument panel for the driver's head. The final design preserved ducting for heating, defrosting, and air conditioning and, except for some interference from the steering wheel-mounted bag system, leaves the driver free to view all instruments.

MAJOR COMPONENTS

Comparison of ESV major components to those in a conventional production vehicle shows increases in weight and complexity in virtually every case. The ESV frame is a prime example of this. Considerable development and several generations of redesign were required to achieve the final configuration shown in Figure 14. The result is a frame of approximately twice the weight of its production counterpart. Complex guide systems inside the convoluted absorber sections had to be developed to insure proper functioning of the energy absorbing system. This design solution is highly experimental and is not feasible for mass production.

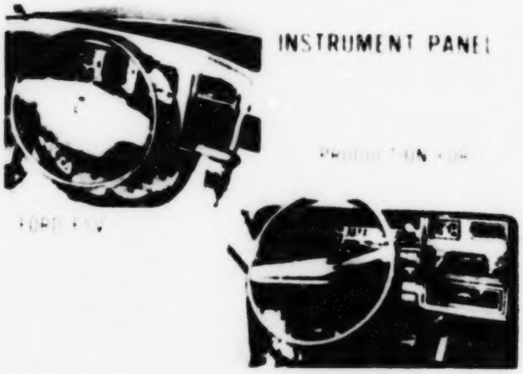


Figure 13

The body front end differs considerably from the production design as can be seen in Figure 15. The addition of the energy-absorbing, corrugated structure causes serious problems of engine compartment packaging and serviceability. As in the case of the frame, body front end design is not suitable for production vehicles.

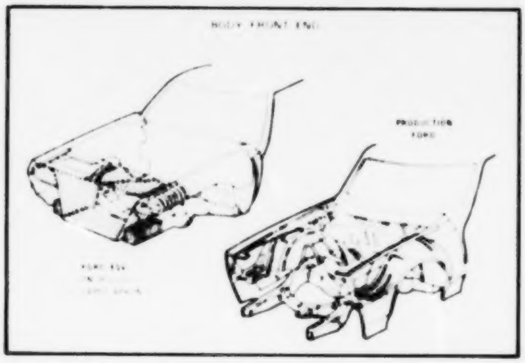


Figure 15

Principal body features are highlighted in Figure 16. They include:

- The structural tunnel between firewall and front seat
- The roll bar at the B-pillar
- The transverse structure between the left and right B-pillar, and
- The double door beams and door latch strikers.

These major items, and many other reinforcements, were necessary to upgrade body integrity, in

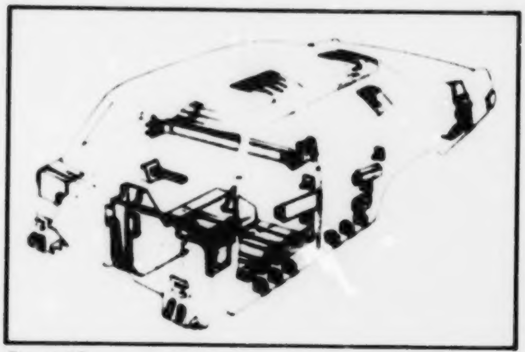


Figure 16

our effort to meet or approach the ESV specified instrusion limits. However, since the basic objective of an intrusion limit is to prevent entrapment of the occupants, we elected to allow more intrusion than specified, wherever doing so would avoid needless structural weight penalties, and as long as occupant g-loads were not increased and the intrusion did not cause entrapment.

Figure 17 shows a current production 5 mph "no-damage" bumper and the ESV 10 mph "no-damage" bumper system.

High strength steel was used for the bumper face bar, so that it incurs minimum plastic deformation during the 50 mph pole crash and transmits some impact forces directly to the frame. This high-strength steel has very limited formability, it was utilized only as an experimental solution to the severe problem of 50 mph pole impact.

Figure 18 compares the current Ford rear bumper system to the ESV rear bumper. Although the rear ESV bumper is very similar in configuration to the front bumper system, the struts are smaller and the face bar is aluminum.

As indicated earlier, the restraint system for the driver (Figure 19) consists of one air bag in the steering wheel, a second bag in the instrument panel,

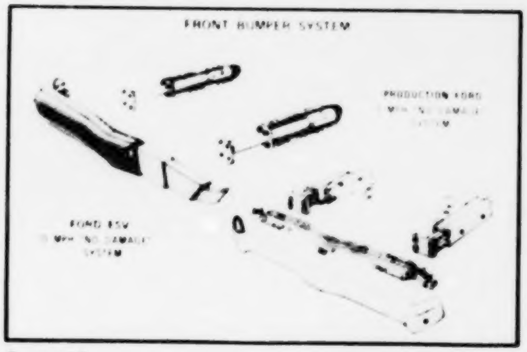


Figure 17

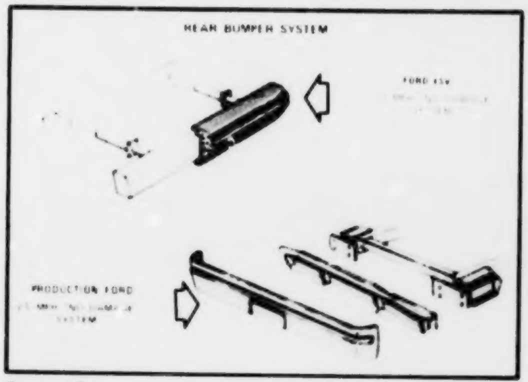


Figure 18

and a controlled-collapse steering column and wheel. A knee impact panel provides additional restraint.

The front and rear passenger restraint systems are very similar, except for the air bag size (Figure 20). One common air bag is used for all three rear passengers in the interest of reducing system complexity. Knee panels are provided to help limit occupant's motion.

The curb weight of our ESV prototype is 5,675 pounds. This represents an increase of 1,375 pounds, or 32%, over a current Ford production model with comparable powertrain and optional equipment. The detail breakdown in Figure 21 shows that most of the weight increase is accounted for by differences in body and frame design and materials.

I would now like to discuss our Crash Test Program.

The following ESV impact tests were conducted at Ford Motor Company's Dearborn Proving Ground:

- 50 mph front barrier
- 50 mph front pole
- 50 mph rear with a moving barrier
- 30 mph ESV-to-ESV side, and
- 15 mph side impact into a fixed pole

All tests were conducted with prototype-design

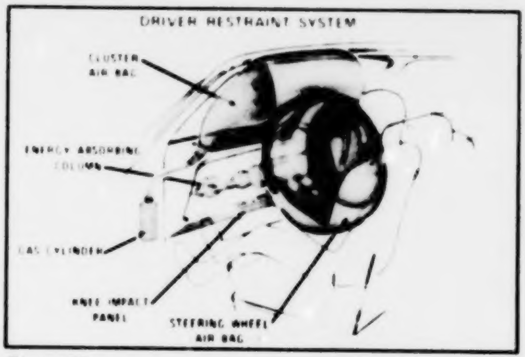


Figure 19

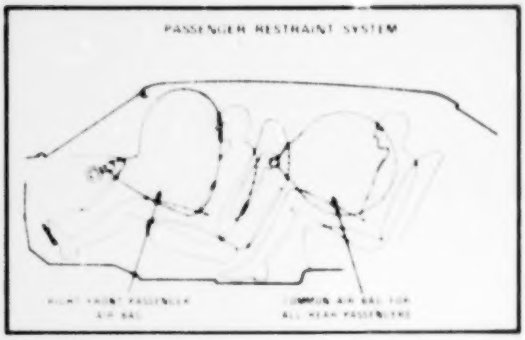

PASSENGER RESTRAINT SYSTEM
RIGHT FRONT PASSENGER AIR BAG
COMMON AIR BAG FOR ALL REAR PASSENGERS

Figure 20

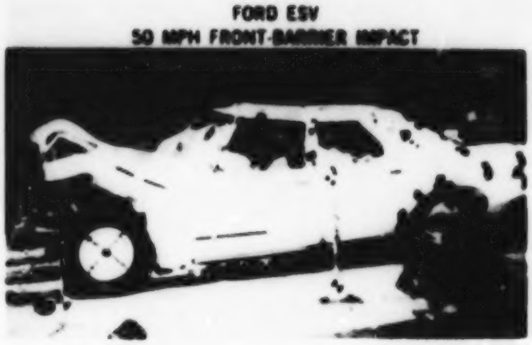
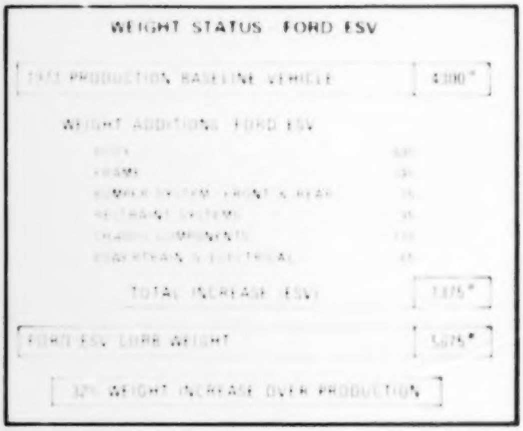


Figure 22



WEIGHT STATUS FORD ESV

1973 PRODUCTION BASELINE VEHICLE	4300#
WEIGHT ADDITIONS FORD ESV	
BODY	[illegible]
FRAME	[illegible]
BUMPER SYSTEM FRONT & REAR	[illegible]
RESTRAINT SYSTEMS	[illegible]
CHASSIS COMPONENTS	[illegible]
POWERTRAIN & ELECTRICAL	[illegible]
TOTAL INCREASE (ESV)	1375#
FORD ESV CURB WEIGHT	5675#

32% WEIGHT INCREASE OVER PRODUCTION

Figure 21

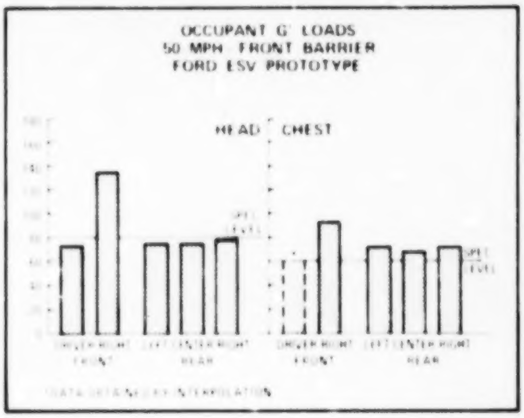


Figure 23

level vehicles, and under the following conditions:

- Sierra 1050 dummies were used throughout our test program.
- All occupants were represented by 50th percentile male dummies except for the center rear position, where a 5th percentile dummy was utilized.

Figure 22 shows an ESV prototype after a 50 mph frontal barrier crash. The energy-absorbing system in the frame and body performed as expected, resulting in 38 inches of static crush, as compared to 36 inches of designed crush. The transmission intrusion into the console structure was six inches; however, there was no evidence of occupant entrapment. Steering gear intrusion to the foot zone was three inches, again with no entrapment.

Figure 23 shows the occupant head and chest g-loads during the 50 mph frontal barrier impact. The channel on driver chest load recorded only intermittently; therefore, the number shown here was obtained by interpolation. The front passenger dummy experienced high g-loads due to air bag system malfunction. Rear occupant *head* g's were within contract specifications, while the *chest* loads were slightly in excess of 60 g's. We must note here that air bag system malfunction was a common problem throughout our test program. Problems were experienced on a random basis and not confined to one particular location or system.

The 50 mph frontal pole impact was by far the most severe test requirement in the ESV program. Figure 24 shows the Ford ESV after the pole test. Note how the hood and the apron structure underneath wrapped around the pole, while the bumper face bar retained its basic shape and transmitted the impact to the frame energy absorbers. The windshield was retained in place, and the static front end crush was 40 inches as compared with the 38 inches projected.

Figure 25 shows occupant "g" loads during the 50 mph pole test. The driver's steering wheel air bag failed to inflate properly in this test, resulting in excessive head and chest loads. Some load readings were lost due to instrumentation failure. Rear occupant head g's were within contract specifications, while the chest loads, once again, exceeded the 60-g limit.

FORD ESV
50 MPH FRONT POLE IMPACT

Figure 24

Figure 26 shows a 4,000 pound moving barrier striking the rear of the ESV at a speed of 50 miles per hour. Note that most of the energy of impact was dissipated by deforming the rear body and frame overhang, with no intrusion into the fuel tank area above the rear axle.

G loads on the front seat occupants during the rear impact test were well within specifications (Figure 27). However, due to head impact to the backlite header, the right rear occupant registered head loads higher than 80 g. The chest g loads on all three rear occupants were above the specified level of 60 g due to bottoming out or complete compression, of the seat back. The seat back was subsequently redesigned to include an added urethane foam panel, thereby providing added energy absorbing capability.

FORD ESV
50 MPH REAR MOVING BARRIER IMPACT

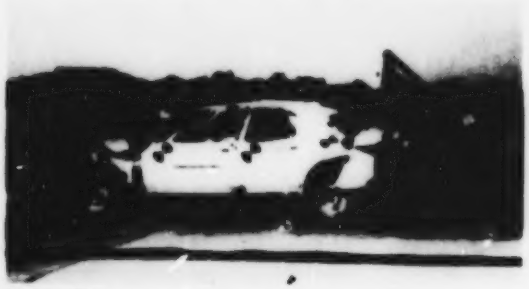


Figure 26

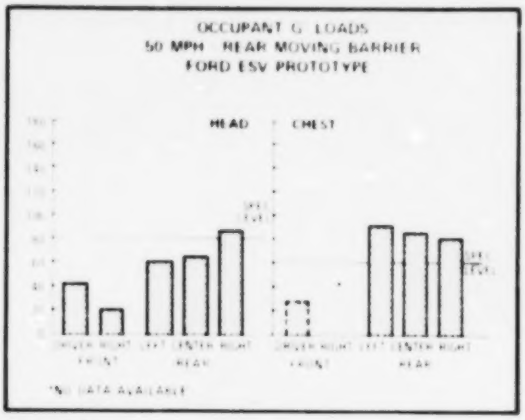


Figure 27

Figure 28 shows the result of striking the ESV with another ESV at 30 miles per hour. The striking car had a front end like the ESV — with a corrugated apron structure, hydraulic struts, bumper system and front frame rail absorbers. On impact, the struts of the striking car fully stroked, and the frame rail absorbers just began to crush. The apron structure was not deformed.

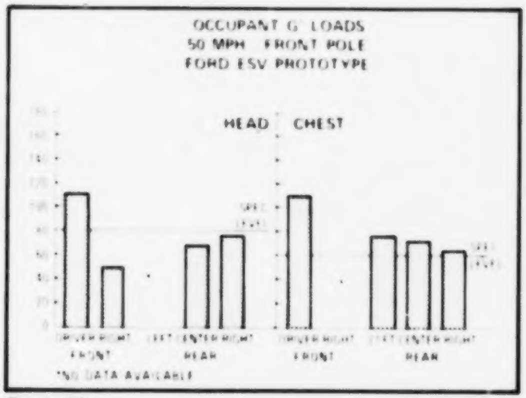


Figure 25

The maximum static interior intrusion into the ESV vehicle was 4.5 inches at the rear door, compared with the three-inch contract specification. Maximum intrusion at the level of the front dummy shoulder and hip or pelvis, however, was only three inches.

FORD ESV
30 MPH SIDE IMPACT—ESV TO ESV

Figure 28

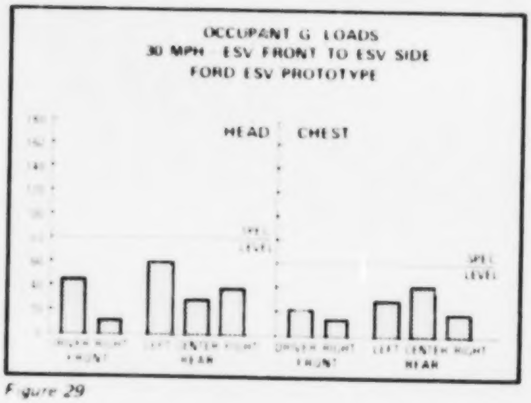
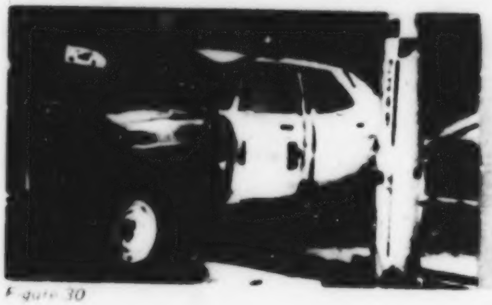


Figure 29



FORD ESV
15 MPH SIDE IMPACT ESV INTO POLE

Figure 30

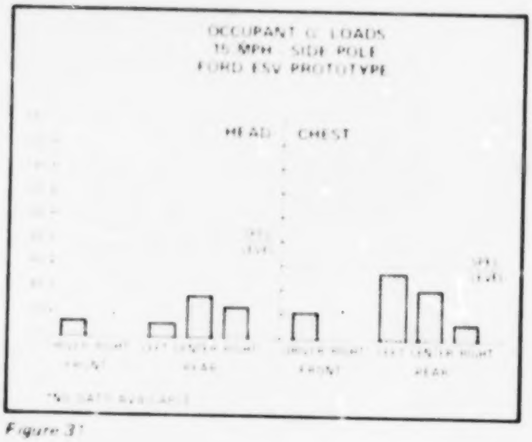


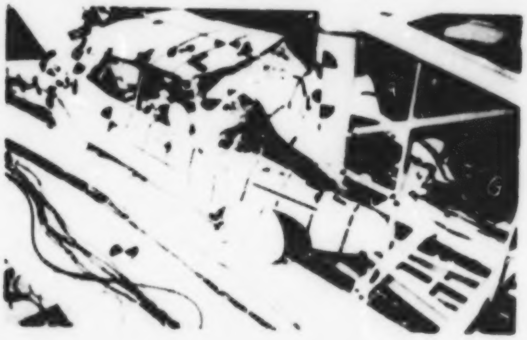
Figure 31

Figure 29 shows the head and chest g's recorded during the 30 mph ESV-to-ESV side impact test. They are all below the 80 and 60 g levels.

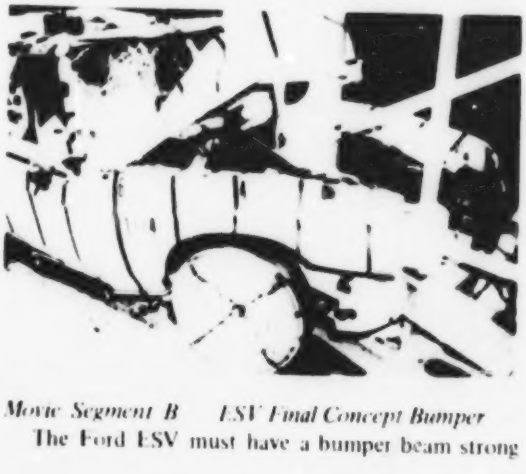
After the 15 mph fixed pole impact, all doors, except for the impacted door, were operational. Maximum intrusion at the center of the impacted door was 7.5 inches, compared to the specification of 4.0 inches. However, intrusion near the dummy's H-point was less than four inches (Figure 30).

Figure 31 shows occupant g loads during the 15 mph fixed-pole test. Even though there was head knocking between the rear center and right occupants, g-levels measured well under the limits specified by the contract. Due to instrumentation failure, no load was recorded for the front passenger dummy. However, past experience indicates that the front passenger normally incurs lower g-loads than the driver.

This concludes the presentation of our crash test results. Following are excerpts from a short film of 50 mph front pole impact crash tests, which demonstrate the difficult task of developing a bumper capable of transmitting the high impact forces to the energy absorbing frame rails without failure.



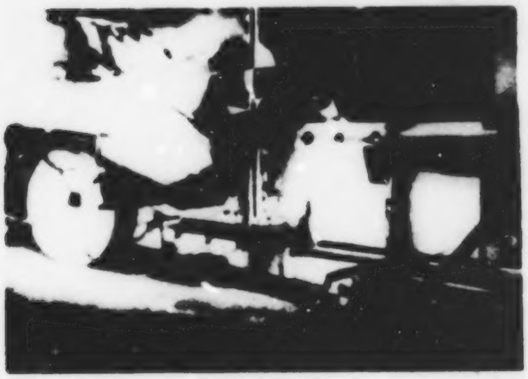
Movie Segment A — 1971 Production Ford — This crash test of a production Ford illustrates the severity of a 50 mph pole impact. Pole intrusion was five feet, resulting in over two feet of engine intrusion through the firewall. A very low level of energy absorption is obtained with this mode of bumper and frame deformation.



Movie Segment B — ESV Final Concept Bumper

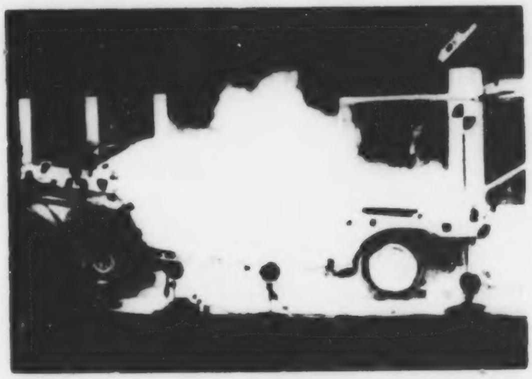
The Ford ESV must have a bumper beam strong enough to compress the frame energy absorbers. Obviously, this first bumper concept did not meet that requirement. The beam fractured on impact as a result of relatively stiff control of our original bumper struts, and the lack of some plastic deformation to reduce the effects of bumper inertia.

Because of the bumper beam failure, the frame rail energy absorbers were not activated at all. The result was excessive pole intrusion of over six feet. The engine was moved more than three feet into the passenger compartment.



Movie Segment C ESV Final Concept Bumper on Prove-Out Vehicle After intensive analysis, scale-model testing and full-size cart tests, a high strength steel bumper design was developed, which is shown here during its first crash test. You can observe the flexing of the bumper, but no bending occurred. The front rail energy absorbers functioned as designed. The side rail energy absorber can be seen under the forward part of the rocker panel. They also compressed properly.

Pole intrusion was now controlled to the projected amount of three feet. Engine intrusion was within specifications. An early version of the controlled collapse apron was installed for this test. It also absorbs a portion of the pole impact energy.



Movie Segment D ESV Final Concept Bumper on Prototype Vehicle A final checkout of the bumper system was conducted on a prototype design level vehicle. This test was also a success. The front frame rail energy absorbers stroked completely. They compressed to approximately 30% of their original length. Side rail absorbers compressed as projected. High bending loads in the side rails required the installation of a guide system inside the convolutions to prevent bind-up. The bumper beam performed with minimal bending. A two-inch thick urethane pad in front simulates the rubber covering of the styled bumper version. Pole intrusion was slightly over three feet. The action of the controlled collapse apron can be observed on either side near the front of the hood.

The data I will show from our crash avoidance performance tests were obtained with our in-house ESV test vehicles on test facilities in Dearborn, Michigan.

Before reviewing the results, I would like to point out that we elected to optimize braking, handling and powertrain performance on the basis of our own engineering judgment, rather than trying to meet the contract specifications in every detail. By so doing, we believe we retained important advantages for our ESV in general public acceptability, reduced complexity and improved reliability.

The Ford ESV brake system is based on production components, except for the hydraulic brake booster, semi-metallic linings for the front disc brakes, and 12-inch diameter rear drum brakes, compared with 11 inches for production. The rear wheel anti-skid system was adopted from the Continental Mark IV.

We were not able to meet the stringent contract requirements for the "full system power on" mode on dry pavement (Figure 32). We approached the objectives at times, but not consistently. The average stopping distance from many test runs was still above the 155 feet requirement. We also exceeded the specification for the "rear breaks only" condition, which assumes front brake failure.

The stopping distance under this failure mode is primarily a function of the vehicle-weight distribution and weight shift during braking. For these reasons, the contract requirement of 343 feet is incompatible with stopping distances obtainable with typical front engine vehicles, such as our ESV.

Figure 33 shows the results of steady-state yaw tests conducted at .4 g and at speeds of 25, 50 and 70 mph. Here, we elected to depart from the specification at 70 mph to provide for more understeer, so as to achieve acceptable handling in lane-change maneuvers, and to improve directional stability under freeway driving conditions.

The transient yaw tests were conducted at speeds of 25 mph and 70 mph. As shown in Figure 34, both tests lie within the limits specified in the contract.

The steering returnability tests were conducted by releasing the steering wheel while cornering at .4 g at 25 and 50 mph. The resultant heading angles shown in Figure 35 are within the specifications. Although the yaw rate is not shown, after two seconds from release time, it was less than four degrees/second at 50 mph, and was zero at 25 mph. Both values met the requirements.

The results of steering control sensitivity tests shown in Figure 36 indicate that the "power-on" condition was above the lower limit specified for all three test speeds of 30, 50 and 70 mph. The "power-off" effort at 70 mph is well below the high limit, whereas efforts at 30 and 50 mph exceeded the upper limit by approximately three inch-pounds. Although our data were above the specified 5 to 1 ratio between "power-off" and "power-on," we

BRAKING RESULTS	
TEST CONDITION	PASSED
FULL SYSTEM – POWER ON	NO
FULL SYSTEM – POWER OFF	YES
FRONT BRAKES ONLY	YES
REAR BRAKES ONLY	NO
EMERGENCY BRAKE – 30% GRADE	YES

Figure 32

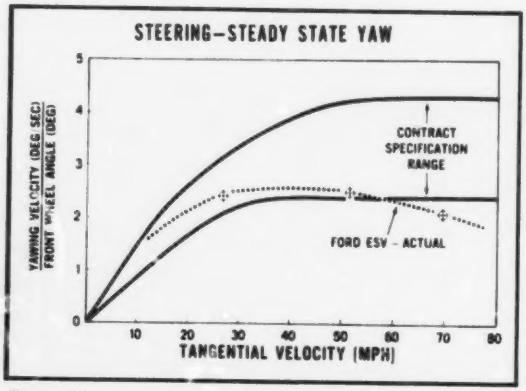


Figure 33

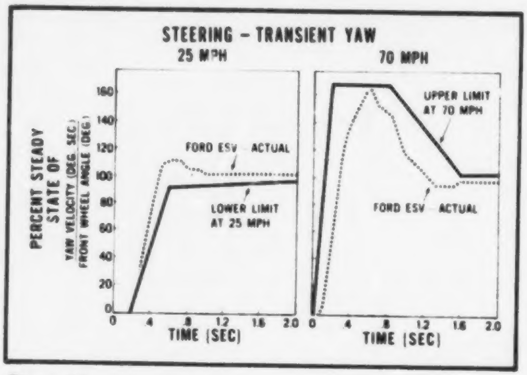


Figure 34

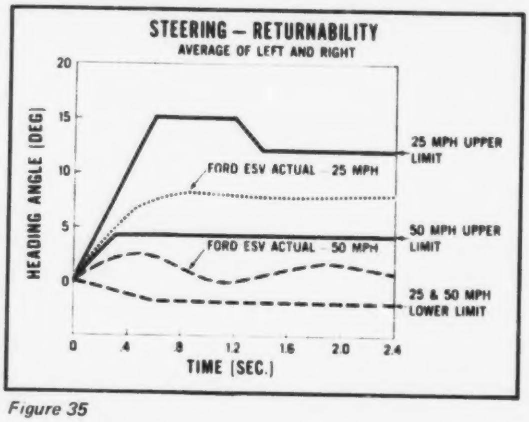


Figure 35

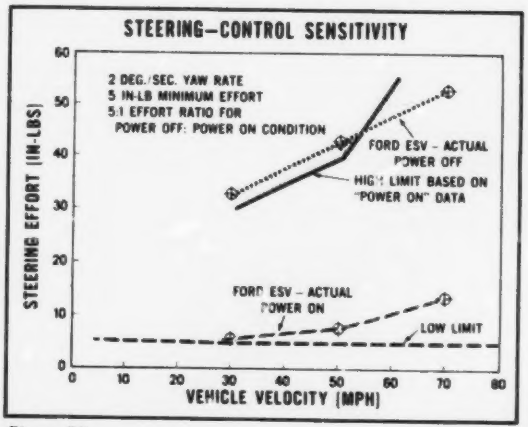


Figure 36

consider the absolute steering effort levels to more realistically reflect acceptable performance.

Figure 37 shows the steady-state, lateral-acceleration test results for the five tire pressure conditions specified. Both the fixed control and manual control test results meet or surpass the specifications.

We are unable to conduct the wet surface tests under the controlled conditions specified in the contract. We have run subjective tests and the results lead us to conclude that we will meet the wet surface tests also. These tests will be conducted in Phoenix on a skid pad with a known wet surface skid number. It should be noted that specially developed experimental tires were used to achieve these results. Their suitability to general use, with regard to wear and other acceptance standards, has yet to be established.

All the ESV powertrain components represent current production. A 351 CID 4V-V8 engine and a three-speed type automatic transmission were used; the rear axle ratio is 3.25. The ESV was able to accelerate from 30 to 70 mph within the specified 12 seconds (Figure 38).

Now I would like to summarize our experience in the ESV program.

- As you have observed from my reports, our original objective to develop systems which could be mass-produced has not been met. In addition, the experimental solutions we had to adopt in many cases, prevented us from estimating the production cost of the ESV, were it produced in volume. In fact, some components were judged not-manufacturable by either known or conceivable production technology.
- After the design and development of our ESV had reached a certain level, we reviewed the overall program to determine if "tradeoffs" against the original specifications should be made. This is a typical procedure which we follow with any of our vehicle engineering programs. We concluded that specification changes in certain areas would be justified by "tradeoffs" for reduced weight and complexity, or for practicality. In our opinion, none of these tradeoffs reduced the overall level of occupant protection afforded by the Ford ESV.
- We have learned much from the ESV program, and hope our effort has contributed useful knowledge and understanding to others involved in ESV activities worldwide.
- We share with most of the other participants in the ESV program concern about the weight and aggressiveness of the ESVs and the high cost inherent in providing extremely high levels of impact resistance. We see the need for more research to determine more realistic specifications.
- In this regard, we have some observations to make in connection with the specifications recently circulated by DOT for the proposed 3,000 pound U.S. ESV. We have given these proposed specifications considerable study and will present our conclusions to you in a brief statement tomorrow morning at the Crashworthiness Seminar.
- Lastly, I would like to announce that Ford Motor Company has agreed to a request by the DOT to add our second ESV prototype to the test program in Phoenix, Arizona. Like our first ESV, this vehicle will be crash tested this summer.

HANDLING – STEADY STATE

LATERAL ACCELERATION – DRY PAVEMENT

TIRE PRESSURE	LATERAL ACCELERATION	
	FIXED CONTROL	MANUAL CONTROL
DESIGN VALUE	PASSED	PASSED
120% DESIGN VALUE	PASSED	PASSED
80% DESIGN VALUE	PASSED	PASSED
120% DESIGN FRONT 80% DESIGN REAR	PASSED	PASSED
80% DESIGN FRONT 120% DESIGN REAR	PASSED	PASSED

Figure 37

ACCELERATION

CONDITION	PASSED	ESV SPEC (SEC'S)	ACTUAL RESULTS (SEC'S')
30 TO 70 MPH PASSING	YES	12.0 OR LESS	11.8

FORD ESV POWER TRAIN

- 351 CID 4V - V8
- C-6 TYPE AUTO. TRANSMISSION
- 3.25 REAR AXLE RATIO
- TIRES MR70 - 15

Figure 38

THE UNITED STATES TECHNICAL PRESENTATION

SECTION TWO

PART 5

RESULTS OF THE OPTIMIZED ESV INTEGRATION AND TRADEOFF STUDIES

ALAN H. ROTH, *Vice President and General Manager*
Advanced Systems Laboratory
American Machine and Foundry, Inc.

Since the last meeting in Washington, AMF has been conducting Tradeoff & Integration Studies for the ESV Family Sedan. This effort has a two-fold purpose. The first is to conduct analyses for the purpose of making recommendations for the new ESV Family Sedan specification. This effort was completed in mid-December, and its results are contained in an interim report to DOT. The second part of this program is to develop improved, crash-injury-reduction design concepts and validate them through development testing. This is currently going on, and we will be completed with this part in June.

Today's presentation will be given by William Rup, ESV Program Manager, and William Wingenbach, our Engineering Director. Mr. Rup will discuss in more detail the objectives of our current program, the scope of the studies, and the status of the program to date. Mr. Wingenbach's presentation will cover several of the significant engineering projects conducted under the scope of our present contract. He will discuss a crashworthiness-weight tradeoff study undertaken to determine the relative impact on weight of the various crashworthiness parameters. This will be followed by a discussion in advancements in our front-end, energy-management system. The effect of the overall advancement in engineering approaches on vehicle configuration and appearance will be given. His presentation will conclude with a discussion of a vehicle simulator which is being developed to assist in the evaluation of the vehicle crashworthiness and aggressivity characteristics.

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SECTION TWO PART 5

THE AMF ESV TRADEOFF AND INTEGRATION STUDIES PROGRAM

WILLIAM RUP, *Program Director*
ESV Programs
American Machine and Foundry, Inc.

The Tradeoff and Integration Studies program, which I will describe (Figure 1), is being conducted by the Advanced Systems Laboratory of AMF, Incorporated under contract with the U.S. Department of Transportation. It has as its objective, the determination of various levels of performance over a range of design conditions, for a number of vehicle subsystems and for the total vehicle. The results of these studies are to be integrated with inputs from the automotive industry, other investigating agencies, and the NHTSA rulemaking agency, to define viable ESV performance levels that are acceptable to all concerned. In addition, the results of this program, which are largely research in nature, are to provide the basis for accomplishing the design of an optimized and production-oriented near-term ESV.

The program is comprised of sixteen studies (Figure 2), which are grouped into seven categories. Three study categories, the interior, front end, and body structure, require development testing in addition to engineering analyses, concept designs and tradeoff evaluations.

The schedule for this program requires that all studies be completed and a final report issued by June 30, 1973. Therefore, at this point in time, I can describe to you the objectives of the studies in each category, the approach taken in conducting the studies, and the current development status of each. We believe that three study areas, front end energy management, the vehicle simulator, and the crashworthiness-weight tradeoff will be of particular interest to the audience; therefore, our next speaker, Mr. Wingenbach, Engineering Director of AMF's Advanced Systems Laboratory, will discuss these studies in detail.

ESV TRADEOFF AND INTEGRATION STUDIES

OBJECTIVE	DETERMINE VARIOUS LEVELS OF PERFORMANCE OVER A RANGE OF DESIGN CONDITIONS
USE	PROVIDE INPUT DATA TO ASSIST NHTSA IN DEFINING VIABLE PERFORMANCE LEVELS
	PROVIDE A BASIS FOR THE DESIGN OF AN OPTIMIZED PRODUCTION ORIENTED ESV

Figure 1

ESV TRADEOFF AND INTEGRATION STUDIES

CATEGORY	STUDY
INTERIOR	PASSIVE FRONT OCCUPANT RESTRAINT SYSTEM REAR OCCUPANT RESTRAINT SYSTEM
FRONT END	ENERGY MANAGEMENT PARAMETRIC VARIATIONS BUMPER CONFIGURATION AGGRESSIVENESS EVALUATION
BODY STRUCTURE	SIDE STRUCTURE REAR END ENERGY MANAGEMENT INTEGRATED STRUCTURE
SYSTEMS	CRASHWORTHINESS WEIGHT TRADEOFF SIMULATION OF VEHICLE DYNAMICS
SIMULATOR	VEHICLE SIMULATOR
SUBSYSTEMS	OCCUPANT RESTRAINT TRADEOFF BRAKES ENGINE EFFECTS ON SAFETY PERFORMANCE REAR VISIBILITY
PRODUCIBILITY	PRODUCIBILITY

Figure 2

PASSIVE, FRONT OCCUPANT RESTRAINT SYSTEM

The interior studies are concerned with two restraint systems, a passive inflatable system for the driver and right front occupant, and a nonpassive belt system for three rear occupants (Figure 3).

For the front system, the specified impact conditions involve on-axis frontal and front-oblique crashes into a fixed barrier. The maximum speed in the on-axis impact is 45 mph, and the impact speed in the oblique impacts varies linearly from 45 mph at 0° to 35 mph at 30°. As a design goal, protection is desired up to 45°, at which angle the speed is 30 mph. The deceleration pulse is trapezoidal in shape, having a rise time of 5 milliseconds to a 29 g plateau. The restraint systems are to accommodate an occupant

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INTERIOR STUDIES
PASSIVE FRONT OCCUPANT
RESTRAINT SYSTEM

IMPACT CONDITIONS:

FRONTAL, FIXED BARRIER, 15 TO 45 MPH

OBLIQUE, FIXED BARRIER, 45 MPH @ 0°
35 MPH @ 30°

PULSE, TRAPEZOIDAL

OCCUPANT RANGE:

5TH PERCENTILE FEMALE

50TH PERCENTILE MALE (PRIORITY SIZE)

95TH PERCENTILE MALE

INJURY CRITERIA:

HEAD	HIC < 1,000
CHEST	60 G's MAX., < 3 MSEC
FEMUR	< 1,400 LBS

Figure 3

size range from a 5th percentile female to a 95th percentile male, with greatest emphasis to be given to the 50th percentile male. The protection to be afforded by the restraint system is defined by the injury criteria shown.

DRIVER'S PASSIVE RESTRAINT SYSTEM

The primary restraint system for both the driver and the right front occupant consists of an inflatable air bag assembly and knee restraints. In addition, the driver's system has a conventional collapsible steering wheel column assembly as a restraining element (Figure 4). Three driver air bag configurations were selected for evaluation testing. Bag volumes, pressure, and fill rates were first established analytically. As testing progressed, these values were modified, as indicated by test data, to obtain improved performance.

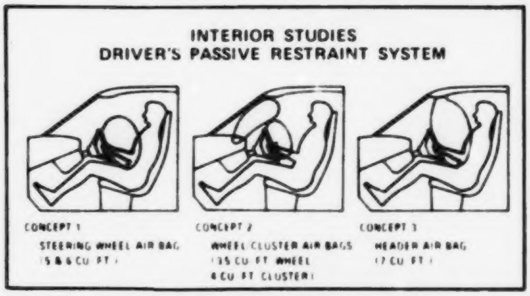


Figure 4

DRIVER'S RESTRAINT SYSTEM, WHEEL/CLUSTER BAG TESTS

The initial series of sizing and evaluation tests have been completed and for the driver the wheel and cluster-bag configuration provided the best performance, as shown by the test data in Figure 5. The parameters that varied in this test series were air-bag characteristics (fill rates and vent sizes) and the steering wheel column angle. The knee restraint characteristics were held constant.

INTERIOR STUDIES
DRIVER'S RESTRAINTS SYSTEM
WHEEL/CLUSTER BAG TESTS

PARAMETER	RUN #998	RUN #999	RUN #1000	RUN #1003
SLED SPEED, MPH	46.7	46.8	46.7	47.1
SLED G'S, MAX.	31.2	31.1	31.7	31.3
DRIVER SIZE	50%M	50%M	50%M	5%F
HEAD HIC	592	380	369	1075
CHEST G'S	74	55	50	60
FEMUR, RIGHT, LBS.	800	1500	1600	1380
FEMUR, LEFT, LBS.	1350	1300	1500	1610
COLUMN STROKE, IN.	6.3	6.5	5.5	0.8

Figure 5

FRONT PASSENGER RESTRAINT SYSTEMS, MODIFIED TEAR DROP BAG TESTS

The right front occupant-restraint evaluation tests were conducted with a 10.3 cu. ft. high-mounted air bag. As with the driver's restraint system, the basic characteristics of the bag were determined analytically and then altered, as dictated by test results. The initial test series provided favorable performance as shown in Figure 6.

A series of qualification tests of the selected driver and occupant restraint systems is now underway. These tests include oblique, as well as on-axis sled

PARAMETER	RUN #998	RUN #999	RUN #1000	RUN #1001	RUN #1002	RUN #1003
SLED SPEED, MPH	46.7	46.8	46.7	46.5	46.9	47.1
SLED G'S, MAX	31.2	31.1	31.7	32.3	32.2	31.3
PASSENGER SIZE	50%M	50%M	50%M	5%F	5%F	95%M
HEAD HIC	665	531	603	365	339	425
CHEST G'S	48	65	55	50	49	55
FEMUR, RIGHT, LBS	1600	1000	1300	2000	1200	1400
FEMUR, LEFT, LBS	1260	1000	980	2000	1550	1450

Figure 6

runs. The preliminary test data indicates restraint system performance equal or superior to the evaluation tests.

REAR OCCUPANT RESTRAINT SYSTEMS

The impact conditions, occupant sizes, and injury criteria to be met by the restraint systems for the rear occupants are the same as those described for the front occupant (Figure 7).

Since a passive system is not required for rear occupants, the restraint concept chosen was the Allied Chemical Corporation's Inflataband System. This restraint system consists of lap and shoulder belts which inflate upon receiving an impact signal and, in effect, provide the belts with a protective action similar to that of an air bag. A series of sled tests is now underway to determine the effectiveness of th Inflatabands in providing rear occupants with the specified protection. Test data from this test series is unavailable at this time.

INTERIOR STUDIES REAR OCCUPANT RESTRAING SYSTEMS	
IMPACT CONDITIONS:	SAME AS FOR FRONT RESTRAINT SYSTEM
OCCUPANT RANGE:	SAME AS FOR FRONT RESTRAINT SYSTEM
INJURY CRITERIA:	SAME AS FOR FRONT RESTRAINT SYSTEM
CONCEPT:	INFLATABAND LAP AND SHOULDER BELTS

Figure 7

FRONT END STUDIES

The front end studies consist of engineering analyses of various energy management systems, the selection of a preferred system and the testing of that system to demonstrate its performance (Figure 8).

Two basic classes of systems were considered. One class, the hybrid system, includes various combinations of hydraulic buffers and crushable structures providing a near-square acceleration pulse which is velocity sensitive. The second class includes fixed-force structures whose resisting force to crash loads increases linearly up to a certain crush distance and then remains constant.

Nine front end energy management systems were investigated, seven of which were hybrid systems with buffer strokes ranging from nine inches to 16 inches and with peak accelerations in a 45 mph flat barrier impact of 30 and 40 g's; the two fixed-force systems had 16 inch and 32 inch ramps to a 40 g peak acceleration.

FRONT END STUDIES	
PURPOSE:	INVESTIGATE VARIOUS ENERGY MANAGEMENT SYSTEMS AND SELECT A PREFERRED SYSTEM DEMONSTRATE FEASIBILITY OF PREFERRED SYSTEM BY APPROPRIATE IMPACT TESTS
CRITERIA:	PASSENGER COMPARTMENT ACCELERATION CRUSH DISTANCE AGGRESSIVENESS PRODUCIBILITY
FINDINGS:	HYBRID ARRANGEMENT WITH 14" STROKE, 32.6 PEAK G AND 27.7 G AVERAGE, SELECTED AS PREFERRED SYSTEM

Figure 8

The criteria used to select a preferred system included the response of an occupant protected by an air bag to the pulse shape produced by each system, the crush distances required by each system, its aggressiveness in impacts with smaller vehicles, and the producibility of each system, including cost and weight considerations.

Based on the analyses conducted to date, a hybrid arrangement with a 14-inch stroke buffer cylinder was selected as the preferred approach. Next month, a bogey test vehicle incorporating this system will be subjected to a series of frontal-barrier and pole impacts and vehicle-to-vehicle crashes to demonstrate the performance of the hybrid approach. The vehicle-to-vehicle tests are designed to evaluate the aggressiveness of the selected hybrid system in impacts with four classes of smaller cars. The descriptions of the front end energy management systems and the vehicle simulator to be used in the aggressiveness testing will be given in the next presentation.

BODY STRUCTURE STUDIES

The body structure category consists of three studies: the first involves the development of a rear-end energy management system; the second, the establishment of the characteristics of the side structure which are optimized in terms of maximum occupant protection and minimum weight; the third study involves the integration of the front end, side and rear-end designs into a structural body configuration, which meets aggressivity and intrusion requirements at a minimum weight (Figure 9). All three studies involve tests on a component level and on a vehicle-system level.

For the rear end system, the maximum allowable passenger compartment acceleration is 40 g's and the

BODY STRUCTURE STUDIES	
PURPOSE	DEVELOP A REAR END ENERGY MANAGEMENT SYSTEM ESTABLISH OPTIMUM CHARACTERISTICS OF THE SIDE STRUCTURE CONDUCT DEMONSTRATION TESTS OF A PREFERRED INTEGRATED STRUCTURE
CRITERIA	COMPARTMENT ACCELERATION AND INTRUSION LIMITS COMPATIBILITY WITH OTHER VEHICLE SYSTEMS PRODUCIBILITY
FINDINGS	CONVENTIONAL DESIGN APPROACHES CAN MEET REAR ENERGY MANAGEMENT REQUIREMENTS MEMBRANE ACTION IN DOOR BEAM AND ROCKER PANEL REQUIRED TO MEET INTRUSION LIMITS

Figure 9

maximum intrusion three inches. The rear end impact conditions are:

- 45° oblique impact with a moving barrier at 25 mph.
- On-axis impact with a moving barrier at 35 mph.
- On-axis impact with an ESV type vehicle at 50 mph.

In addition, there is a no-damage criterion to be met in a 5-mph impact with a moving barrier.

For the side structure, the crash conditions include a rigid seven inch diameter pole impact with the center of a side door at 15 mph, and ESV-type vehicles impacting the side structure normally and obliquely. A four-inch maximum permanent intrusion in the pole impact and a three-inch maximum is allowable in the vehicle impact. A design goal for the side structure is to limit the peak passenger compartment acceleration to 20 g's.

The impact conditions and the structural response levels specified for the individual subsystems also apply to the integrated structure.

Other criteria that apply to the body structure studies require the structure to be compatible with the vehicle's functional requirements, such as occupant seating, suspensions, drivelines, etc., and that the structure be producible in terms of conforming with industry practices that are projected for the late 1970's.

The engineering analysis required by these studies was conducted using finite element techniques. From a design standpoint, the rear-impact mode is least critical. The studies indicate that a rear-end-structure failing in simple Euler buckling, followed by crippling of sheet metal components, furnishes sufficient energy absorption and that the 30 g passenger compartment acceleration is feasible using relatively conventional structural design approaches. Tests of a representative rear end structure design just concluded verify the analytical findings.

In the side structure study, the analyses indicate that the design goal of 20 g's peak lateral passenger compartment acceleration in the pole impact is feasible with an appropriately designed door beam, rocker panel and supporting structure. For this impact mode, membrane action must be designed into the door beam and rocker panel to provide the required resistance to intrusion. Since the crush stroke is limited, this membrane action must be developed as soon as possible after the instant of impact. This can be accomplished using honeycomb inserts and/or foamed plastic-filled cores for the door beam and the rocker panel.

Several door beam designs are now undergoing tests in order to develop the best approach.

VEHICLE SIMULATOR FOR CRASH TESTING

I had indicated previously that one of the studies of the front end energy management system involves tests to determine the system's aggressiveness in frontal impacts with other vehicles. The purpose of the vehicle simulator is to provide a testing device which will simulate vehicles having a weight range from 1,500 pounds to 3,500 pounds and a fixed-force type of front structure, and is to be used in determining the behavior of the ESV structural test vehicle in frontal impacts (Figure 10).

The simulator has the capacity to absorb the energy and force levels shown, which correspond to the 3,500 pound simulator configuration and an impact speed of 45 mph. The simulator is capable of being towed up to 60 mph and of having its gross weight adjusted to 5,000 pounds. Its front end characteristics can be varied by programming the desired force/stroke response.

VEHICLE SIMULATOR FOR CRASH TESTING	
PURPOSE	PROVIDE A TEST VEHICLE SIMULATING VARIOUS FRONT END RESPONSES IN FRONTAL IMPACTS
REQUIREMENTS	GVW VARIABLE FROM 1500 TO 3500 LBS 238,000 FOOT POUNDS MAXIMUM ENERGY 160,000 LBS MAXIMUM FORCE TOWABLE UP TO 60 MPH

Figure 10

MAJOR SYSTEMS STUDIES CRASHWORTHINESS-WEIGHT TRADEOFF

The crashworthiness-weight tradeoff study is an analytical study relating vehicle weight and weight distribution to specified crash requirements. The results of the study include a set of relationships between crash parameters and vehicle weight for one basic vehicle design and a set of performance requirements, and the criteria for evaluating particular sets of crash requirements (Figure 11).

SYSTEM STUDIES
CRASHWORTHINESS WEIGHT TRADEOFF

PURPOSE	ESTABLISH THE RELATIONSHIPS BETWEEN CRASH REQUIREMENTS AND RESULTANT VEHICLE WEIGHT
STATUS	ANALYTICAL MODEL USING MATRIX NOTATION DEVELOPED • FIVE CRASH CONDITIONS CONSIDERED • CRASH CONDITIONS TREATED INDIVIDUALLY OR IN COMBINATIONS • OUTPUT PLOTS WEIGHT INCREMENT AS A FUNCTION OF IMPACT VELOCITY ON A SUBSYSTEM OR VEHICLE LEVEL

Figure 11

In conducting this study, an analytical mode representative of component-level weight relationships has been developed. The model consists of an assemblage of masses coupled to each other and to the crash-parameter velocities squared through an energy-coefficient coupling matrix and a basic weight matrix. The assemblage of masses can then be operated on by a geometric coefficient matrix to obtain the total vehicle weight and weight distribution. The five crash conditions considered are front flat barrier, front pole, side pole, vehicle-into-vehicle side, and rear-moving barrier. The crash conditions can be treated individually or in combinations.

The computer output plots weight as a function of impact velocity for the user-requested crash condition. The weight data can be plotted on a subsystem or total vehicle level. This study will be discussed in more detail by the next speaker.

VEHICLE DYNAMICS MATH SIMULATION

The second systems study involves vehicle dynamics (Figure 12). The purpose of this study is to determine the effects on vehicle dynamic performance produced by varying the weight and weight distribution on an ESV-type automobile. The weight range considered in this study is from 3,500 to 6,000 pounds.

In conducting this study, an existing vehicle simulation computer program, which has been verified using ESV dynamic performance test data, is utilized. The input parameters include descriptors of the vehicle mass, vehicle-mass distribution, vehicle geometry as well as the characteristics of the suspension, steering and tire systems. Specific values for the mass and mass-distribution data is obtained from the crashworthiness-weight tradeoff study.

SYSTEM STUDIES
VEHICLE DYNAMICS SIMULATION

PURPOSE	DETERMINE EFFECTS ON VEHICLE DYNAMICS PRODUCED BY VARYING WEIGHT AND WEIGHT DISTRIBUTION • VEHICLE WEIGHT RANGE 3500 TO 6000 POUNDS
STATUS	ANALYTICAL MODEL USING DIGITAL COMPUTER SIMULATION SELECTED

Figure 12

The primary output of interest is tabulated and plotted data of the vehicle transient yaw response. Steady state yaw response is obtained from the output of several runs, each run having a different velocity input. Secondary output is tabulated and presents data on deflections, velocities, accelerations and forces for the various vehicle components of interest.

OCCUPANT RESTRAINT TRADEOFF

The occupant restraint tradeoff study involved the investigation of the merits of currently developed restraints and the recommendation of a preferred occupant restraint system for the projected ESV configuration. Thirteen types of restraint concepts were considered, these included passive and non-passive systems, inflatable systems in air bag and belt configurations, and blankets.

For preliminary evaluations, the candidate restraint concepts were examined for their effectiveness and cost at each individual seat position. The measure of effectiveness was based on the *Injury Criteria Model for Restraint System Effectiveness Evaluation* developed by the Highway Research Safety Institute and considers the five criteria shown. The most promising combinations of the individual seat position restraints were then evaluated to arrive at the preferred restraint system for the vehicle. The final evaluations of the vehicle restraint combinations used a number of additional criteria including probable occupant acceptance, reliability, potential hazards, and effect on vehicle design.

BRAKING PERFORMANCE ANALYSIS

The second of the subsystem studies (Figure 13) involves the analysis of the various brake system configurations to determine their relative per-

SUBSYSTEM STUDIES
BRAKING PERFORMANCE ANALYSIS

PURPOSE	DETERMINE BRAKING PERFORMANCE OF VARIOUS BRAKE SYSTEM CONFIGURATIONS
BRAKING CONDITIONS	STRAIGHT LINE STOP FROM 60 MPH
	STOP FROM 30 MPH IN A 0.3 G LATERAL ACCELERATION TURN
	ROAD TIRE FRICTION μ = [illegible]
STATUS	ANALYTICAL MODEL USING HYBRID COMPUTER SIMULATION SELECTED

Figure 13

formance in straight-line and accident-avoidance type of braking. The study involves a comparison of six brake systems. A foundation brake system, consisting of conventional front-wheel disc brakes and rear-wheel drum brakes, with fixed-load proportioning between the front and rear wheels, provides the standard for performance comparisons. This brake system configuration is modified to provide five other systems by the addition of variable load proportioning between the front and rear wheels, variable side-to-side load proportioning, rear wheel anti-skid features with fixed and variable front-to-rear load proportioning, and four-wheel anti-skid features. The braking conditions consider straight-line stopping and stopping-while-in-a-turning-maneuver for various vehicle weights.

This analysis, currently underway, utilizes an existing computer program to provide data on stopping distances versus pedal force, wheel forces, vehicle trajectory and yaw gain in turning maneuvers, and other performance data as a function of time. Braking performance under two failure modes, front brake-line and booster-type failures, is also being investigated.

ENGINE EFFECTS ON SAFETY PERFORMANCE

The engine effects study (Figure 14) investigated current engine developments, selected those engines that will probably be used in production vehicles in the late 1970's and evaluated them in terms of their effect on safety performance. The basic specifications required an engine of approximately 300 horsepower and capable of meeting the 1975 emission standards. Candidate engines or prime movers considered in this study include Rankine, Stirling, electric, gas turbine, the rotary Wankel and the conventional reciprocating V-8.

Of these, only the Wankel and the conventional engine were judged to meet the production criterion and can be considered as candidate engines for use in a near-term ESV. The Wankel engine, because of its smaller size and lower weight, provides the potential of improved safety performance over a conventional engine. Some of the improvements that could be realized in safety vehicle designs are increased crush distance and volume for energy management systems, improved forward-down-angle visibility, a decreased intrusion threat to the firewall and a lower weight.

SUBSYSTEM STUDIES
ENGINE EFFECTS ON
SAFETY PERFORMANCE

PURPOSE	EVALUATE ENGINE DEVELOPMENTS AND THEIR EFFECT ON SAFETY PERFORMANCE
CRITERIA	IN PRODUCTION IN LATE 1970's
FINDINGS	ROTARY ENGINE (Wankel) IS A CANDIDATE FOR NEAR TERM USE ESV
	PERMITS PERFORMANCE IMPROVEMENTS OVER CONVENTIONAL ENGINES

Figure 14

REAR VISIBILITY TRADEOFF

The fourth subsystem study is concerned with investigating various rearviewing systems and other means of obtaining improved rear visibility in a near-term safety vehicle (Figure 15). Some of the specific topics that are addressed in the study include the attainment of the proposed FMVSS 111a requirements for indirect visibility without the use of a periscope, and the feasibility of eliminating the C-pillar without degrading the vehicle's structural crashworthiness.

The indirect-viewing systems that were studied included electro-optical devices such as television and

SUBSYSTEM STUDIES
REAR VISIBILITY TRADEOFF

PURPOSE	INVESTIGATE VARIOUS REAR VIEWING SYSTEMS TO OBTAIN IMPROVED REAR VISIBILITY
CRITERIA	COST EFFECTIVENESS
FINDINGS	PLANAR MIRROR SYSTEMS ARE PREFERRED

Figure 15

fiber optics, optical devices such as planar and convex mirrors, and various types of periscopes. The prime criterion for selecting a preferred rearviewing device is cost effectiveness. Consideration is also being given to the impact a rearviewing system will have on the overall vehicle design. Of the indirect rearviewing devices considered, the planar mirror systems were judged to be most cost effective. However, the study found that the visibility standard proposed for implementation in 1976 could not be met with planar mirrors in a vehicle of conventional design.

The feasibility of eliminating the C-pillar as a means of enhancing rear visibility is now being analyzed as part of the body structure studies. Upon completion of this work, any permissible changes in the C-pillar area will be established, and a determination made of the resulting indirect field of view.

I had indicated that producibility is an evaluation criterion in many of the tradeoff and integration studies. The method used in determining producibility is to examine each design concept for conformance with production materials and methods currently in use or projected to be used in the late 1970's. Nonconforming design concepts are modified or alternative designs developed to attain acceptable producibility levels.

SECTION **TWO** PART **5**

CRASHWORTHINESS-WEIGHT TRADEOFF STUDY

WILLIAM WINGENBACH, *Director*
Engineering Department
American Machine and Foundry, Inc.

I will discuss several of the tasks which we have been conducting in support of an optimized family sedan ESV program.

The first of these tasks is a crashworthiness-weight tradeoff study (Figure 1). The discussion will cover the task objective, the analytical model used as the basis of computer program, crash conditions used to design the baseline vehicle, and the results obtained. The basic objective of this study was to provide a meaningful set of relationships between crashworthiness parameters and the vehicle system and subsystem weights.

The analytical model was developed using matrix notation for both ease of computer manipulation and physical insight into the problem (Figure 2). Briefly, the model consists of an assemblage of masses coupled to each other and to the crashworthiness parameters through an energy-coefficient coupling matrix and a basic functional-weight matrix. The assemblage of masses is operated upon by a geometrical-coefficient matrix to obtain total vehicle weight, as well as other mass properties that may be required. The matrix relationship resulting from this description is shown.

The (M_V) matrix contains the system-level mass and inertia properties. The (G) matrix contains user-defined geometric transformations such as summation terms and subsystem CG locations. The (M_O) matrix contains terms specifying the functional weight for each subsystem (or component). The (E_C) matrix contains the energy coupling coefficients relating impact velocity and the elements of the (R) matrix to subsystem and component weight variation. The expression within the outer brackets represents the collection of subsystem and component weights.

The baseline OESV has been designed to meet performance requirements for the set of crash conditions shown in Figure 3. For each crash condition shown, the combination of velocity, impact position or impact angle resulting in the worse-case design condition was used for each element.

The analytical techniques and computer methods previously mentioned have been applied to the OESV design. The vehicle has been divided into crashworthy

CRASHWORTHINESS WEIGHT TRADEOFF STUDY

INTRODUCTION, OBJECTIVES
- ● ANALYTICAL MODEL AND METHODS
- ● CRASH CONDITION REQUIREMENTS
- ● RESULTS
- ● SUMMARY

Figure 1

METHOD OF ANALYSIS

MATRIX METHODS EMPLOYED

BASIC EQUATION

$$[M_V] = [G] \quad [M_O] + [E_C] [R]$$

WHERE		
	M_V	OVERALL VEHICLE MASS
	G	GEOMETRY TRANSFORMATIONS
	M_O	BASIC MASS OF ASSEMBLIES WITH NO CRASH CRITERIA IMPOSED
	E_C	COUPLING COEFFICIENTS FOR MASS ENERGY
	R	CRASH REQUIREMENTS

Figure 2

CRASH CONDITIONS

IMPACT AREA	CRASH CONDITION	VELOCITY MPH (KPH)	IMPACT ANGLE DEG	CENTERLINE IMPACT POSITION
FRONT	FIXED BARRIER	[illegible]	0 – 45	
	FIXED POLE	40 (64.4)	0	℄ – W/4 TO ℄ + W/4
SIDE	FIXED POLE	15 (24.1)	90	A TO C PILLAR
	CAR	30 (48.3)	45 – 135	℄ FRONT SEAT H POSITION
REAR	MOVING BARRIER	[illegible]	135 – 180	℄ REAR WHEELS

Figure 3

Preceding page blank

and purely functional subsystems and components. The crashworthy subsystems include the front and rear energy-management systems and the integrated structural elements, while the functional subsystems include suspension systems, engine, drive train, etc. The resultant vehicle weight for the baseline set of crash conditions shown is 4,678 pounds. This consists of a functional weight of 4,223 pounds and crashworthy structure weight addition of 455 pounds. The functional weight is defined as that amount of weight required for each component for purely functional requirements, that is, with all crash condition requirements removed.

The next four figures show vehicle curb weight variation obtained from crashworthiness parameter variations.

Figure 4 shows the variation of vehicle curb weight for the front barrier and front pole crash conditions. For each crash condition, two subcases have been run. These are the baseline case, where each crash condition not being varied is held at its baseline value, and the suppressed case, where each crash condition not being varied is suppressed by setting its magnitude to zero. These two types of runs allow the effects of individual crash conditions to be investigated both with their relationship to other crash conditions and also without the influence or interaction of the other crash conditions.

Note the difference in minimum weight obtained for the two sets of curves shown. The baseline case, minimum weights, reflecting the weight requirements of all other crash conditions kept at their baseline velocities, are 4,616 pounds for the barrier impact, and 4,627 pounds for the pole impact. The suppressed case functional weight for both requirements, when the impact velocity is reduced below 27 mph, is the vehicle functional weight of 4,223 pounds. The difference of approximately 400 pounds for each case reflects the weight required for nonvaried crash conditions. The effect of a simultaneous variation of both frontal requirements will be shown later.

Similar data for the side impact requirements is shown in Figure 5. Since both the side-pole and side-vehicle crash conditions affect, primarily, the same structural elements, very little weight variation results from varying a single side-impact crash condition. In addition, many of the load carrying elements in side impacts are also significant longitudinal load carrying elements for the front and rear crash conditions. The effect of simultaneous variation of these two conditions is shown in Figure 6.

Figure 6 summarizes the results obtained for the seven baseline parametric cases run, including the two that consider the simultaneous variation of the two front and two side-impact velocities. The rear barrier crash conditions case is also shown. A significant item

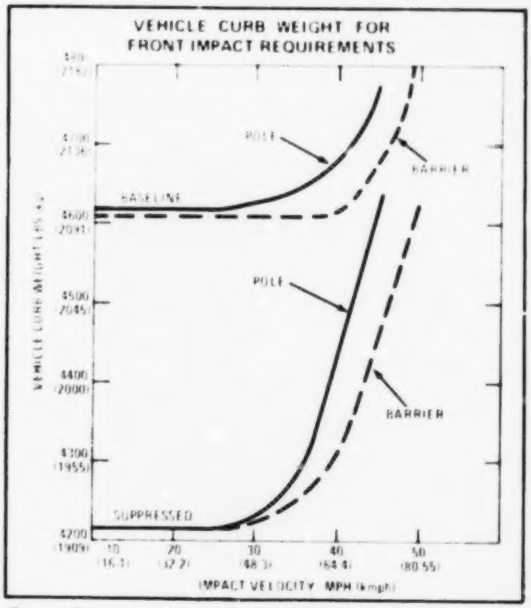


Figure 4

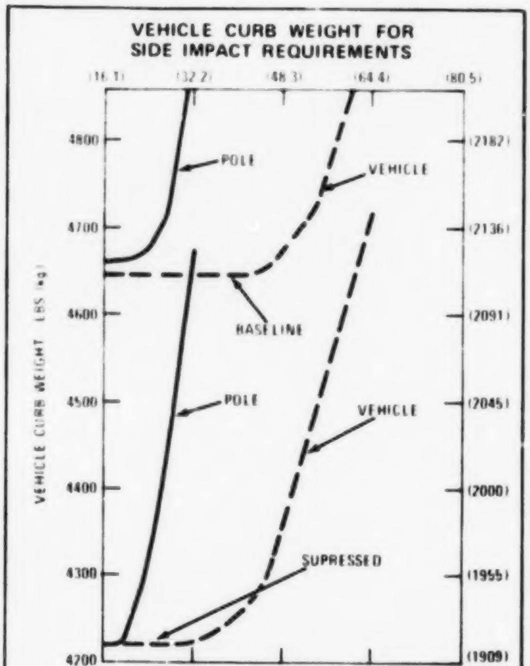


Figure 5

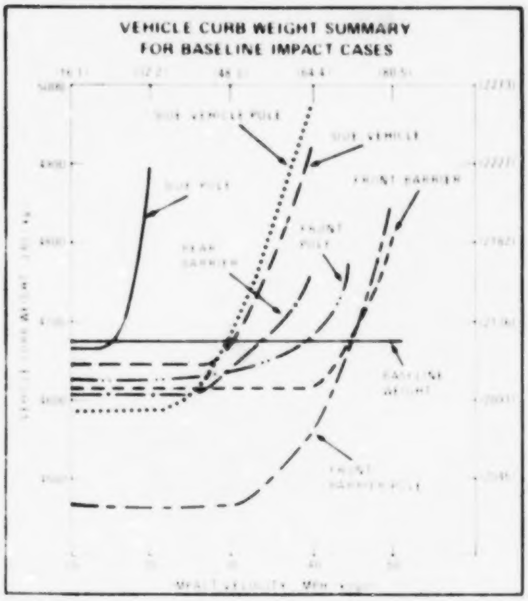


Figure 6

is the difference in resultant weight obtained when the front barrier and pole crash conditions are varied simultaneously by maintaining a constant 5 mph difference in impact requirements. An additional weight reduction of 157 pounds is indicated at frontal impact velocities of 30 and 25 mph for the barrier and pole respectively, versus either reduced individually. This difference is reduced, of course, as the baseline velocities are approached. The same trends are noted for the combined side vehicle/pole variation, although not as dramatic as the front crash conditions.

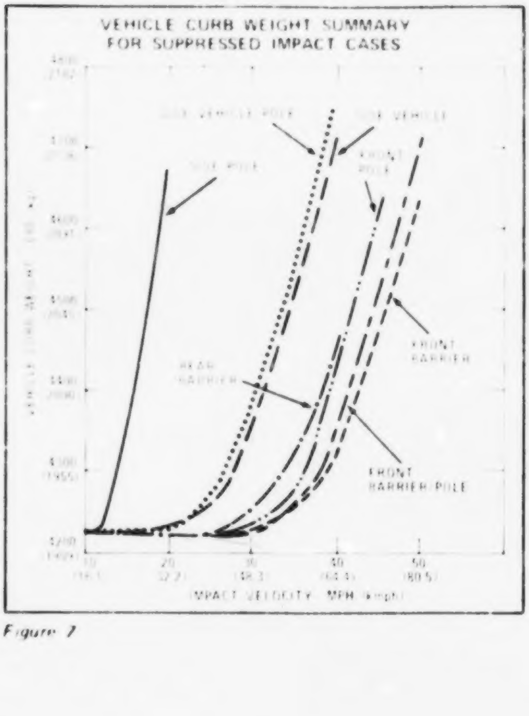


Figure 7

A similar summary for the seven suppressed cases is shown in Figure 7. The data shown here compares the weight resulting from the variation of individual crash condition requirements independently.

In summary, AMF has developed and demonstrated a system-level, user-oriented, analytical tool that has been employed to gain insight into complex crashworthiness and weight interrelationships for our OESV design. Full utilization of this tool will be obtained when the effects on weight of crashworthiness parameter variations are related to expected benefits from these crashworthiness parameters in terms of crash injury reduction.

FRONT ENERGY MANAGEMENT SUBSYSTEM

I have been discussing the effect of crashworthiness parameters on the total vehicle system. One of the important subsystems contributing to vehicle crashworthiness is the front energy-management subsystem.

The prototype AMF ESV utilized variable stroke hydraulic buffers to absorb virtually all the required energy under all impact conditions (Figure 8). These units had a maximum stroke of 30 inches and were mounted rigidly in the vehicle frame. Oblique and off-center impacts were handled with the help of a sophisticated bumper-connection linkage which allowed the buffer pistons to remain parallel while stroking unequal distances, and, with the help of a method for modulating the buffer resistance, on the basis of the total load exerted by both buffers. This latter device made the acceleration pulse relatively insensitive to the obliquity or amount of offset of the impact.

This system, although it provided excellent performance under a wide variety of impact conditions had two disadvantages: it was heavy, weighing over 500 pounds and it was complex in terms of fabrication and installation. It was apparent, as the system was being developed, that the only way in which significant weight savings could be achieved would be to allow the vehicle structure to share in the energy-absorption function.

This, then, forms the background for the current development effort on the front energy-management system. The goal of this effort is to produce a system which, while possessing the nonaggressive characteristics of a velocity sensitive hydraulic system, is

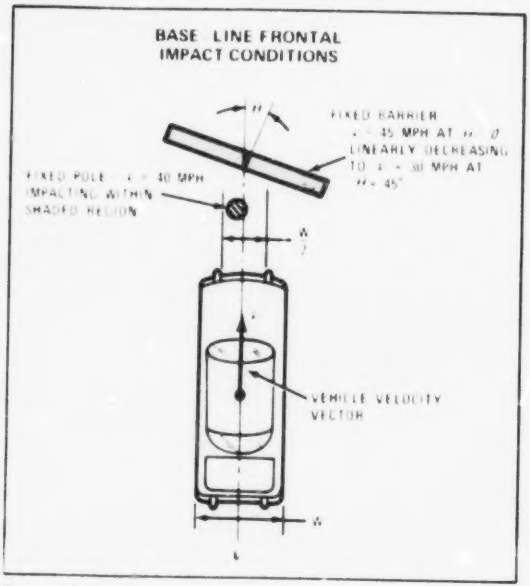

BASE LINE FRONTAL
IMPACT CONDITIONS
FIXED BARRIER
V = 45 MPH AT θ = 0
LINEARLY DECREASING
TO V = 30 MPH AT
θ = 45°
FIXED POLE V = 40 MPH
IMPACTING WITHIN
SHADED REGION
W/2
VEHICLE VELOCITY
VECTOR
W

Figure 8

also light in weight and relatively simple. The system is being developed to meet the baseline impact requirements as shown.

The development is being carried out in three phases. The first of these was a trade study and evaluation based on preliminary design work. The second phase was a detailed design of a test version of a preferred system. The final phase, which is now underway, includes the fabrication and testing of this system.

The system which was studied in the first phase was a hybrid using a combination of hydraulics and deformable structure. Hydraulic systems varying in stroke length from nine inches to 16 inches were considered. The primary conclusions drawn were the following. First, for a given hydraulic unit stroke, a system providing more front end crush distance and lower passenger compartment acceleration is not only superior in terms of performance to a shorter, stiffer front end, but is also marginally lighter. That is, the lighter structure required, in both the front end and passenger compartment, compensates for the extra length of sheet metal required. Second, for a "soft" system, the weight penalty associated with using a 16-inch stroke hydraulic unit instead of a nine-inch stroke is only about 28 pounds, provided that high-strength steel is used in both the cylinder barrel and piston. This appeared to be a modest penalty in view of the advantage of the longer stroke in terms of aggressiveness and occupant protection in low speed impacts.

As a result of these conclusions, the philosophy used in the detailed design of the new system was to make the front end as soft as possible and the hydraulic stroke as long as possible (within a 16-inch limitation), while meeting the baseline impact requirements and remaining within the basic package limitations set by a 220-inch overall vehicle length and the use of a standard V-8 engine for power.

Figure 9 is a schematic of the variable stroke hydraulic buffer. It is conceptually the same design as that used on the original AMF ESV. The major differences are a shorter (14 inch stroke versus 30 inch) and a self-contained accumulator, as opposed to the external accumulator system used previously.

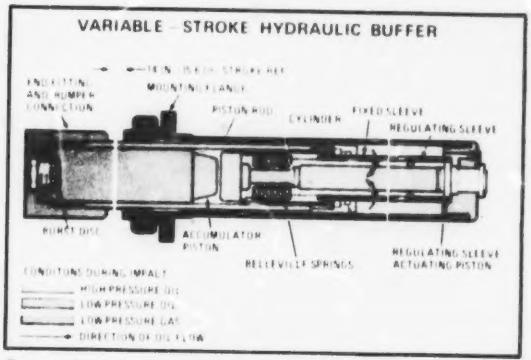

VARIABLE – STROKE HYDRAULIC BUFFER
14 IN. (35.6 CM) STROKE REF
END FITTING
AND BUMPER
CONNECTION
MOUNTING FLANGE
PISTON ROD
CYLINDER
FIXED SLEEVE
REGULATING SLEEVE
BURST DISC
ACCUMULATOR
PISTON
BELLEVILLE SPRINGS
REGULATING SLEEVE
ACTUATING PISTON
CONDITIONS DURING IMPACT
HIGH PRESSURE OIL
LOW PRESSURE OIL
LOW PRESSURE GAS
DIRECTION OF OIL FLOW

Figure 9

A plan view of the test version of the front end is shown in Figure 10. The longitudinal elements are parallel to the ground, at bumper height (17 inches). This arrangement gives a maximum available crush of about 34 inches, with the bumper solid against the engine and the engine solid against the firewall. The hydraulic buffers nest inside the crushed frame members and the completely compressed buffer is just about the same length as the engine. Although only 30 inches of crush is expected in the 45 mph

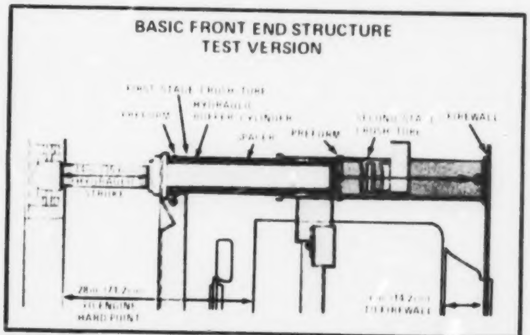

BASIC FRONT END STRUCTURE
TEST VERSION
FIRST STAGE CRUSH TUBE
HYDRAULIC
BUFFER CYLINDER
PREFORM
SPACER
PREFORM
SECOND STAGE
CRUSH TUBE
FIREWALL
HYDRAULIC
STROKE
28 in (71.2 cm)
TO ENGINE
HARD POINT
TO FIREWALL

Figure 10

barrier impact, it is estimated that about 40 inches will be needed in the offset pole impact. This will result in local compartment intrusion of six inches.

One of the major problems in designing such a hybrid, hydraulic and structural, energy-absorption system is to integrate the two modes of energy absorption so that they complement each other in both 0° and oblique impacts. As indicated in Figure 10, there are two stages of structural crush in addition to the hydraulic unit. Both of these are cylindrical tubes of mild steel, the first being 4.5 inches O.D. with a .109-inch wall, and the second, 5-inch O.D. with a .083-inch wall. Note that both of these tubes are preformed to ensure that failure begins at the desired location.

During a high speed impact, the hydraulic unit strokes simultaneously with crush of the first stage tube. A spacer is provided between the hydraulic cylinder and tube, which provides support to the tube and limits the extent of the region in which the tube is free to crush. Any vertical or transverse loads and moments are carried out by the spacer into the portion of the tube which is not free to crush. Thus, such loads do not have a significant effect on the longitudinal load generated by the tube.

The engine mounts, which are boxed in to restrain the engine in all directions, and the front suspension are carried primarily between the first and second stage tubes. Thus, the inertia load on the first stage tube is about 1,000 pounds more than on the second stage. This fact, together with the tube size selection ensures that crush of the first stage will be completed before a significant amount of crush of the second stage occurs. Consequently, the crush of the second stage tube does not begin until the hydraulic cylinder and spacer are already inside it, and it crushes around the spacer.

The resultant system is a variable stroke hydraulic unit with a maximum stroke of 14 inches. In a 45 mph, 0° barrier impact, an additional 16 inches of structural crush occurs giving a total crush of 30 inches. The acceleration and crush characteristics, as functions of impact velocity in a 0° barrier impact, are shown in Figure 11 for both the new design and the prototype ESV. Six inches of stroke are provided for impacts below 5 mph; above this speed, the hydraulic stroke increases to its maximum value of 14 inches at 15 mph, and remains there until an impact velocity of about 30 mph is reached at which point structural crush begins.

Figure 12 shows a summary of the component weights for the new energy-management system along with those for the prototype ESV system. The numbers indicate that we have achieved the goals of simplifying and lightening the front energy management system. The weight of the total front energy management has been reduced from 705 pounds for the original ESV to 359 pounds for the optimized ESV.

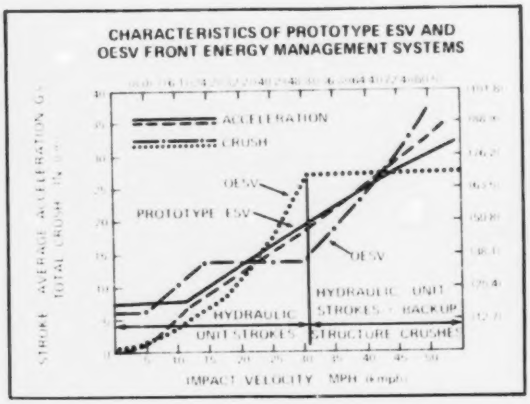


Figure 11

WEIGHTS – FRONT ENERGY MANAGEMENT AND STRUCTURE FORWARD OF FIREWALL		
DESCRIPTION	WEIGHT (dry) – lb (kg)	
	ESV	OESV
BUMPER	124 (56.3)	70 (31.8)
BUFFERS WITH BUMPER CONNECTION HARDWARE	355 (161.1)	130 (59.0)
PLUMBING	20 (9.1)	—
ACCUMULATORS	26 (11.8)	—
BASIC VEHICLE STRUCTURE FORWARD OF FIREWALL (Includes Crush Tubes in OESV)	180 (81.7)	159 (72.2)
TOTALS	705 (320.0)	359 (163.0)

Figure 12

ORIGINAL ESV VERSUS OPTIMIZED ESV

There are several other important differences between the original AMF ESV and the optimized version. This original was fabricated around a high-strength steel space frame and had a very husky appearance. The vehicle had a fiberglass exterior which did not contribute to impact energy absorption.

The optimized version will be fabricated in a manner which is representative of techniques used on production vehicles. Figure 13 shows the construction of the inner body. This will be fabricated from conventional automotive steel and will be covered

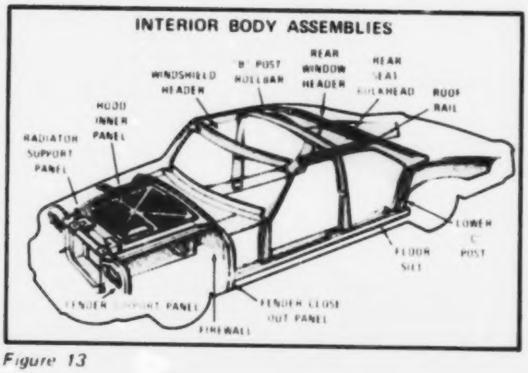

INTERIOR BODY ASSEMBLIES
WINDSHIELD HEADER
"B" POST ROLLBAR
REAR WINDOW HEADER
REAR SEAT BULKHEAD
ROOF RAIL
HOOD INNER PANEL
RADIATOR SUPPORT PANEL
LOWER "C" POST
FLOOR SILL
FENDER SUPPORT PANEL
FENDER CLOSE OUT PANEL
FIREWALL

Figure 13

with an all-steel exterior. This resulting assembly is a conventional unitized body.

Figure 14 shows the crashworthy structural assembly. I have previously discussed the front energy-management system. The vehicle floor and side structure are used as a load path for front and rear impact loads. The door panels are similar to the crushable beam membrane units used on the original although they are now mounted below the window glass and will be more effective in impacts by vehicles into the side. The rear-end energy management is accomplished by structural deformation of the rear sills and supporting structure. The entire vehicle is fabricated of conventional automotive materials, with the exception of the front bumper which is aluminum, the front buffers, which are high-strength steel; and the door panels, which are aluminum honeycomb. All elements of the vehicle are considered to be mass-producible, as designed, with the exception of the front buffers, which will require further development.

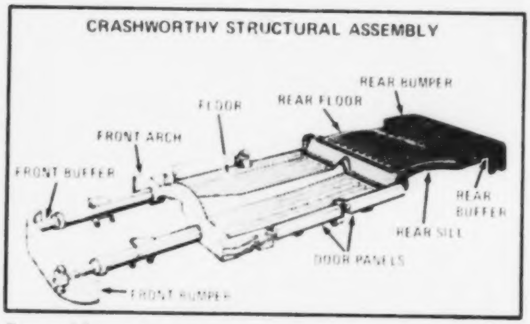

CRASHWORTHY STRUCTURAL ASSEMBLY
REAR BUMPER
FLOOR
REAR FLOOR
FRONT ARCH
FRONT BUFFER
REAR BUFFER
REAR SILL
DOOR PANELS
FRONT BUMPER

Figure 14

VEHICLE SIMULATOR

An important concern in the development of the ESV has been the control of its aggressiveness when impacted against other vehicles. We have been developing a vehicle simulator which will enable us to test any vehicle against another vehicle of arbitrary weight and crush characteristics. The objectives of the vehicle simulator are shown in Figure 15.

VEHICLE SIMULATOR OBJECTIVES

1. ASSIST IN EVALUATION OF AGGRESSIVITY CHARACTERISTICS OF EXPERIMENTAL SAFETY VEHICLE
2. SIMULATE ARBITRARY VEHICLE FRONT END CHARACTERISTICS
3. SIMULATE VEHICLES WEIGHTS 1500 LB TO 5000 LB (681 kg to 2270 kg)
4. TOWABLE TO VELOCITY UP TO 50 MPH (80.5 km/h)
5. ENERGY ABSORPTION CAPACITY UP TO 250,000 FT LB (34,575 kg meter)
6. DECELERATION FORCE UP TO 160,000 LB (45,400 kg)
7. DECELERATION STROKE UP TO 24 IN. (61 cm)

Figure 15

The deceleration force is provided by a programmable, servo-controlled, hydraulically-actuated braking system. The method of operation is to measure force at the bumper and to measure stroke of the bumper. These values are compared to the desired force-stroke relationships. The braking force is then adjusted as required. This process is continuous during the deceleration stroke.

The vehicle shown in Figure 16 consists of a chassis and a moving-bumper assembly. The chassis consists of a frame, running gear and brake applicator, while the bumper assembly consists of a bumper, force-measuring system and brake bars. The simulator is towed at the desired speed and impacts with

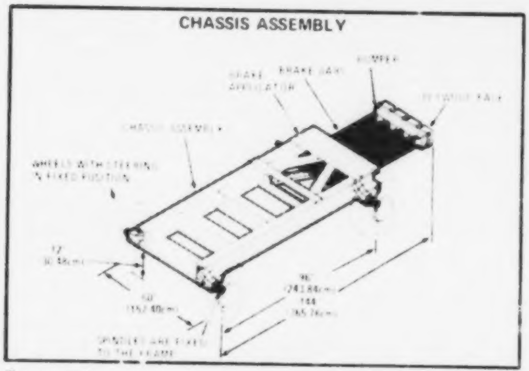

CHASSIS ASSEMBLY
BRAKE APPLICATOR
BRAKE BARS
BUMPER
CHASSIS ASSEMBLY
WHEELS WITH STEERING IN FIXED POSITION
12"
(30.48cm)
60"
(152.40cm)
96"
(243.84cm)
144"
(365.76cm)
SPINDLES ARE FIXED TO THE FRAME

Figure 16

another vehicle. The moving element strokes axially, while force is generated by friction between brake pads and the brake bars. There is a plywood face on the bumper which serves to reduce the initial impact force on the bumper. This face is replaced after each test. This is the only refurbishment required between tests. The force measurement system consists of eight load cells located behind the bumper.

Figure 17 shows the front view of the brake applicator assembly. There are four, vertically-oriented brake bars connected by a thin plate. This is the moving part of the brake assembly. Force is applied hydraulically by a series of pistons acting on brake pads which in turn generate the braking force. One of the most serious problems encountered in the design of a programmable servo-controlled braking system is to get the system to respond quickly and accurately. It is necessary to keep the fluid compressibility and brake structure deformation to a minimum. The design shown is very compact, both hydraulically and structurally. Computer simulations indicate that proper resultant system response will be achieved.

Figure 18 shows the general arrangement of hydraulic components. Energy for operating the system is stored in the high-pressure accumulator, which is pressurized to about 3,000 psi prior to a mission. Servo valves are mounted directly to the brake applicator. The detachable unit to the left provides ground support in the form of electrical power and high-pressure hydraulic supply for pre-mission preparation. This is disconnected prior to a test.

Figure 19 shows the simulator electrical subsystems. Of particular interest is the position indicator. This is a combination optical-electrical system in which light pulses are generated as the bumper assembly strokes. These light pulses are converted to an electrical signal which is fed to the force programmer. This signal, along with the electrical signal from the load cells, is interpreted in the servo system. If the braking force is different from that required, the servo valves are actuated and the braking force is adjusted. An electronic control module is located at the rear of the vehicle which is at the left side of Figure 19. This module contains the control and vehicle checkout system. Included as plug-in replaceable modules are the force program cards.

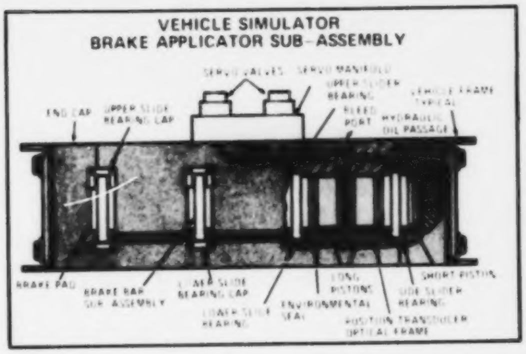

VEHICLE SIMULATOR
BRAKE APPLICATOR SUB-ASSEMBLY
SERVO VALVES
SERVO MANIFOLD
UPPER SLIDER BEARING
VEHICLE FRAME TYPICAL
END CAP
UPPER SLIDE BEARING CAP
BLEED PORT
HYDRAULIC OIL PASSAGE
BRAKE PAD
BRAKE BAR SUB-ASSEMBLY
LOWER SLIDE BEARING CAP
LOWER SLIDE BEARING
ENVIRONMENTAL SEAL
LONG PISTONS
POSITION TRANSDUCER OPTICAL FRAME
SIDE SLIDER BEARING
SHORT PISTON

Figure 17

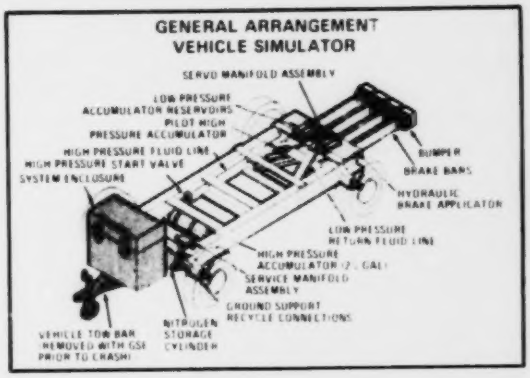

GENERAL ARRANGEMENT
VEHICLE SIMULATOR
SERVO MANIFOLD ASSEMBLY
LOW PRESSURE ACCUMULATOR RESERVOIRS
PILOT HIGH PRESSURE ACCUMULATOR
HIGH PRESSURE FLUID LINE
HIGH PRESSURE START VALVE
SYSTEM ENCLOSURE
BUMPER
BRAKE BARS
HYDRAULIC BRAKE APPLICATOR
LOW PRESSURE RETURN FLUID LINE
HIGH PRESSURE ACCUMULATOR (2 GAL)
SERVICE MANIFOLD ASSEMBLY
GROUND SUPPORT RECYCLE CONNECTIONS
NITROGEN STORAGE CYLINDER
VEHICLE TOW BAR (REMOVED WITH GSE PRIOR TO CRASH)

Figure 18

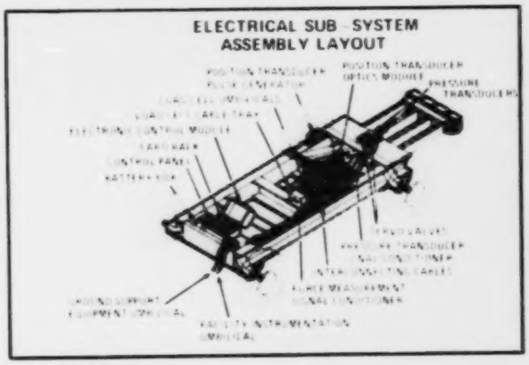

ELECTRICAL SUB-SYSTEM
ASSEMBLY LAYOUT
POSITION TRANSDUCER PULSE GENERATOR
POSITION TRANSDUCER OPTICS MODULE
PRESSURE TRANSDUCERS
LOAD CELL UMBILICALS
LOAD CELL CABLE TRAY
ELECTRONIC CONTROL MODULE
CARD RACK
CONTROL PANEL
BATTERY BOX
SERVO VALVES
PRESSURE TRANSDUCER SIGNAL CONDITIONER
INTERCONNECTING CABLES
FORCE MEASUREMENT SIGNAL CONDITIONER
GROUND SUPPORT EQUIPMENT UMBILICAL
FACILITY INSTRUMENTATION UMBILICAL

Figure 19

A card for a typical force program is shown in Figure 20. The only limitation at present on the characteristics of the force-deflection profile, in addition to previously mentioned limits on force and energy, are that the profile can be adequately described by six straight lines. This is an arbitrary limitation and could be increased if necessary.

Figure 21 shows the computer-predicted response of the simulator during an impact, controlled by the force program shown. The initial spike is a function of the mass of the moving element and the stroke of the plywood facing material. This system is capable of following a force change on the order of 20 kips per millisecond with a steady state error of about 20 kips. As can be seen, the response of the simulator to

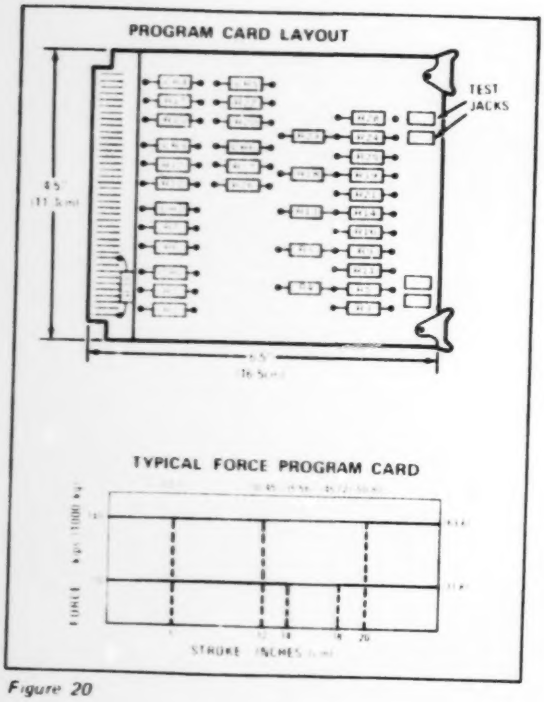


Figure 20

a fairly complex force-deflection profile is quite good. The status of the simulator is that it is currently being assembled and checked out for performance. This work is expected to be completed by the end of the month. The simulator will be available for test purposes in April of this year.

CONCLUSION

ALAN ROTH, *Vice President and General Manager*

You have just seen our current effort on ESV trade studies as well as our structural-design concept for the optimized vehicle. We've been pleasantly surprised during the past year at the re-test capability of the AMF ESV, beyond expectations in impacts with barriers, with current production vehicles, and most recently, with another ESV of a small weight range. The vehicle has served its purpose and has performed admirably. In the next phase of the program, we will concentrate on achieving a marked improvement in producibility without sacrificing safety features.

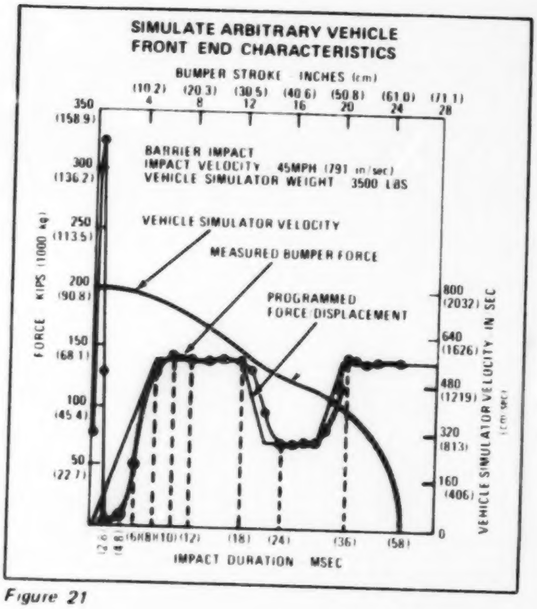


Figure 21

It is a requirement of the optimization phase of this program to demonstrate high safety performance in a vehicle built utilizing the same materials and techniques that are used in current production automobiles. This requirement will be met and with it, the original intent of the program, to support the generation of future standards, will be accomplished.

THE UNITED STATES TECHNICAL PRESENTATION

SECTION TWO PART 5

THE OPTIMIZED ESV PERFORMANCE SPECIFICATIONS

WILLIAM E. SCOTT, *Acting Director*
Office of ESV Programs
National Highway Traffic Safety Administration

It is my pleasure to review for you the preliminary specifications/goals for a U.S. 3,000 pound class ESV. I say review, because most of you have already received the specifications, and I hope had an opportunity to study them in detail. In any case, I will not attempt to cover every aspect of the specifications, but instead will discuss the major performance criteria and changes from our original "family sedan" ESV requirements.

These preliminary specifications are not to be taken as our final ideas of the necessary or complete performance requirements for this class vehicle. Rather, we ask you to regard them as goals toward which our efforts should be directed and whose attainment we believe will contribute to significant advances in automobile safety.

I regret that time will not permit detailed discussion of these performance goals this morning. But, I hasten to add that tomorrow morning's parallel sessions on Accident Avoidance and Crashworthiness are specifically set aside to provide an opportunity for comment and discussion.

The ESV program has, from its inception, planned to address four weight classes of experimental safety vehicles that would span the vehicle mix anticipated on our highways. The first effort was logically a 4,000 pound family sedan, to be followed by a 3,000 pound weight class ESV. You will recall that this plan was presented to the international community at the first ESV conference in Paris, France (Figure 1).

The U.S. 3,000 pound ESV will be an integrated Research Safety Vehicle which will be used to assess the effectiveness, practicality, and achievability of safety design concepts that could be incorporated into mass-produced vehicles of the 1980's. We believe this to be a logical follow-on to the U.S. family sedan prototype and optimized projects which will influence vehicle designs that could be mass produced in the late 1970's.

In selecting this vehicle, major consideration was given to projected changes in the vehicle mix anticipated on our nation's highways in this and the next decade, as well as energy, resource and pollution problems. We will not commit to building vehicles in this weight class until we have had the opportunity to receive and consider comments from all of you.

Briefly, the plan for proceeding with the development of this project is as follows:

1. We will initiate, within a month or two, a phase one conceptual design study. We expect our phase one work to build upon existing work in this class by participants in the International ESV Program.
2. At the same time, we will increase our R&D efforts in structures, restraints and vehicle handling.
3. Approximately one year later, we will initiate a preliminary design effort based upon the results of the phase-one study, Subsystems R&D and Automotive Industry Inputs.
4. Final design and fabrication of a 3,000 pound weight class ESV will follow completion of the preliminary design studies.

The major point is that commitment to hardware is at least two years away and in the interim, recognizing your current ESV efforts, we ask you to join with us in developing the final specifications for this class vehicle.

U S 3000 POUND ESV PROJECT

BACKGROUND

- RESEARCH VEHICLE AIMED AT 1980'S
- LOGICAL FOLLOW ON TO U S FAMILY SEDAN PROJECT
- RESPONSIVE TO VEHICLE MIX ENERGY POLLUTION AND RESOURCE PROBLEMS
- COMMITMENT TO HARDWARE WILL AWAIT INDUSTRY INPUTS AND FURTHER STUDY

Figure 1

3000 POUND PROJECT OBJECTIVES

- APPLY KNOWLEDGE FROM FAMILY SEDAN ESV TO NEW CLASS OF VEHICLE
- EMPHASIZE CONFORMITY WITH AUTOMOTIVE PRODUCTION PROCESSES AND MATERIALS
- INCREASE EMPHASIS ON VEHICLE AGGRESSIVENESS AND PEDESTRIAN SAFETY
- PROVIDE INPUT TO FMVSS'S FOR 1980'S

Figure 2

We hope to attain four primary objectives during the course of this program (Figure 2).

- First, we plan to use the knowledge gained from the family sedan prototype, the international effort, and optimized projects as a basis for advancing the state-of-the-art in automobile safety design in this class.
- Second, we plan to make sure that the design of the 3,000 pound ESV is in keeping with anticipated production techniques and materials.
- Third, in contrast to the family sedan specification, we plan to give much more attention to vehicle aggressiveness and pedestrian safety.
- And last, we plan to use data derived from this program as a major input to the development of Federal Motor Vehicle Safety Standards for the 1980's.

The basic requirements for the U.S. 3,000 pound ESV are summarized in Figure 3. Relative to the prototype family sedan specification, the target weight has been decreased 1,000 pounds. The wheelbase has been decreased 18 inches, the length has been decreased 35 inches, and the height has been decreased two inches. The minimum number of passengers has been reduced from five to four. Driver and front passenger size requirements are unchanged, but the minimum acceptable rear-passenger size has been reduced from 95th percentile to 50th percentile male. Design options are allowed for engine/driveline layout, number of doors, and cargo space arrangements.

BASIC VEHICLE REQUIREMENTS

- **GENERAL CONFIGURATION**
  - WEIGHT 1360 KG/3000 LBS (MAX.)
  - WHEELBASE 2700 MM/106 IN (MAX.)
  - LENGTH 4700 MM/185 IN (MAX.)
  - HEIGHT 1420 MM/56 IN (MAX.)
- **PASSENGER COMPARTMENT**
  - FOUR PASSENGERS (MIN.)
  - DRIVER SIZE THRU 95TH MALE
  - PASSENGER SIZE THRU 50TH MALE
- **DESIGN OPTIONS**
  - ENGINE/DRIVELINE LAYOUT
  - DOORS (2 OR 4)
  - CARGO SPACE ARRANGEMENT

Figure 3

Now I would like to give you a brief description of the more significant safety performance requirements reflected in this specification.

SAFETY
HANDLING AND STABILITY

- **BRAKING**
  - STOPPING DISTANCE 175 FT/55 M FROM 60 MPH
  - REAR WHEEL ANTI LOCK
  - BRAKING IN A TURN
- **STEERING**
  - STEADY STATE YAW RESPONSE INCREASED RANGE TO NEUTRAL STEER
  - CLARIFICATION AND ALTERATION OF TRANSIENT YAW RESPONSE

Figure 4

First, in the area of accident avoidance, a number of changes have been made in braking requirements (Figure 4). Stopping distance on dry surfaces for the 3,000 pound ESV has been raised to 175 feet, which is 20 feet longer than that required for the prototype family sedan vehicle. Although this is largely a relaxing of requirements, the vehicle will be tested fully loaded, rather than at 40 percent load as required for the first ESV prototypes. However, 175 feet is 19 feet less than required in the current Federal Motor Vehicle Safety Standard 105a, which specifies 194 feet for the same test conditions.

Included in the braking specification are two other major changes. One requires that, as a minimum, anti-skid brakes be provided for the rear axle of the vehicle. The other change specifies brake performance in a combined braking and steering maneuver.

The anti-skid system, as well as a number of other braking system configurations, is being evaluated by AMF as a part of their ESV Tradeoff and Integration Studies. Even though this evaluation is not complete, we believe that two-wheel anti-skid systems have reached a state of development that makes their mass production costs acceptable.

The braking, in a turn maneuver, is based largely on the results of the University of Michigan, HSRI study that was published in January of this year and is titled *Vehicle Handling Performance*. The requirements of the specification are derived from the 12 production cars that were evaluated in that test program, and we believe them to be modest for a vehicle with a well designed anti-skid brake system. Certainly the safety relevance of this maneuver is more straightforward than some others may be, since a significant fraction of drivers will instinctively attempt to steer and brake when an obstacle presents itself. Incidentally, in the specifications sent to you on February 16, 1973, Figure 1, showing "Vehicle Deceleration versus Brake Pedal Force" and Figure 2, "Steady State Yaw Response versus Tangential Velocity" were in error. Errata sheets have been sent and are available here.

The steering requirements contained in this new specification are also somewhat different from those in the family sedan specifications. For steady state yaw response (Figure 5), we have extended the acceptable range up to the neutral steer line and opened up the lower boundary by allowing more under-steer beyond 50 mph. These alterations were based on: 1) the performance data we obtained from the four ESVs tested in the past year, 2) comments gathered during visits to auto manufacturers, and 3) the fact that, currently, we have no data to justify the original envelope, except to state that it describes our domestically-produced vehicles. Further, you will notice that the response parameter plotted on the ordinate has been normalized to take into account the variable effects of vehicle wheelbase. This modification will allow comparisons between vehicles to be made more readily.

The transient yaw response curve (Figure 6) has been altered slightly to clarify acceptable performance and to allow for vehicle response lag to steering input, based on our definition of Time Zero. As described in the specification text, the area below the upper line is acceptable to 70 mph and the area above the lower line is acceptable to 25 mph. Therefore, the total acceptable area is dependent on test speed. The area enclosed by the two lines is merely common to both speeds. To accomplish the allowed lag in yaw response for the 25 mph test, the first leg of the envelope has been shifted to the right. It now intersects the time axis at .2 seconds rather than 0.0 seconds. This adjustment is consistent with our test procedure when "t" zero is defined as the point in time at which the total steering input is 50 percent complete.

For the handling portion of the specification (Figure 7), the only significant change has been the deletion of the manually-controlled lateral acceleration tests. This was done for two reasons: 1) the results are consistently very close to fixed control tests, and 2) we would like to eliminate driver variability, wherever possible, from the test procedures.

In the area of overturning immunity, we have decided, based on the results of our ESV tests conducted over the past year, to raise the speed through the pylon course from 45 mph to 50 mph. The 45 mph requirement was easily attained by ESVs with wheelbases up to 124 inches. Therefore, 50 mph was thought to be more realistic for the performance of a 100-inch wheelbase car.

Although this vehicle is smaller than our family sedan, we anticipate no great difficulties in meeting all of the original engine and driveline requirements. Any problems encountered in meeting range and lateral force influence on engine performance requirements are secondary, relative to meeting the passing time requirements.

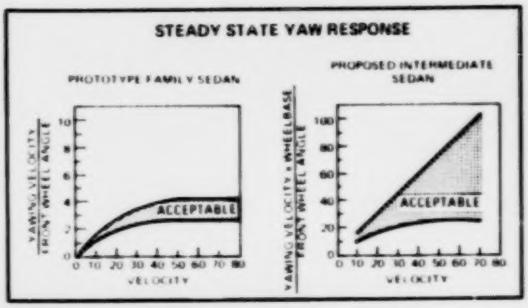


Figure 5

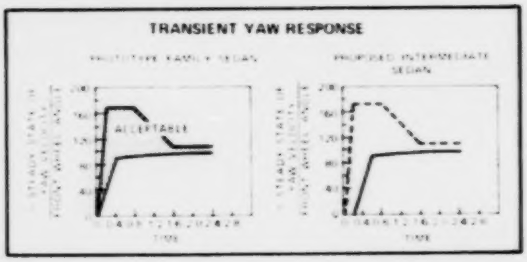


Figure 6

SAFETY
HANDLING AND STABILITY

- HANDLING
  DELETION OF MANUAL CONTROL LATERAL ACCELERATION TEST
- OVERTURNING
  PYLON COURSE SPEED INCREASED TO 50 MPH/80 KPH
  J- TURN MANEUVER REPLACED BY MORE DRASTIC STEER AND BRAKE MANEUVER
- ENGINE AND DRIVELINE PERFORMANCE
  NO CHANGE

Figure 7

In our attempt to define ESV requirements in a manner more closely related to rulemaking, the driver-visibility requirements are based upon the proposed standards, prototype ESV data, and docket comments (Figure 8). For the direct field of view we have adopted plane surfaces, which reduce design efforts as limits rather than conical surfaces. We have attempted to resolve potential conflicts with requirements for rollover strength and head restraints. Therefore, Zone I is modified from 17° left and right of the X Plane to 15° left and right of the X Plane. Zones II and III are expanded accordingly and the allowable horizontal width of obstructions in these zones is increased. The steering wheel, windshield

SAFETY
VISIBILITY

- **DRIVER VISIBILITY**
  - BASED ON PROPOSED STANDARD AND DOCKET COMMENTS
  - PERISCOPE NOT REQUIRED
  - BACKLIGHT (REAR WINDOW) WIPER REQUIRED
- **LIGHTING**
  - BASED ON PROPOSED STANDARD AND DOCKET COMMENTS
  - VERTICAL AIM ADJUSTMENT, DUAL INTENSITY SIGNALS AND LAMP FAILURE INDICATORS DESIRED

Figure 8

SAFETY
DRIVER ENVIRONMENT

- CONTROLS AND DISPLAYS
  - SPECIFIC INSTRUMENTATION REQUIRED (MIN.)
  - ONE "G" OMNIDIRECTIONAL RESTRAINT FOR DRIVERS
- WARNING DEVICES
  - AUDIBLE SPEED WARNING DEVICE
  - PASSIVE RESTRAINT CONDITION MONITORING
  - MECHANICAL FAILURE WARNINGS DESIRED
- ENVIRONMENT
  - CARBON MONOXIDE LEVELS
  - SUBJECTIVE COMFORT LEVELS
  - AIR CONDITIONING REQUIRED
- EMERGENCY EQUIPMENT

Figure 9

wipers and head restraints shall not be considered as obstructions.

We feel our indirect field of view requirements are the minimum acceptable for cars of the early 1980's. Although we have purposely avoided requiring a periscope, we have *not* prohibited one. We have found that periscopes are very costly and cause systems-integration problems with structures and restraints. We have also allowed a convex mirror on the right (passenger) side.

The backlight (rear window) wiper requirement is included as an effort to improve visibility to the rear under adverse conditions.

In the case of lighting, we adopted the proposed lighting standard as the minimum requirement, since this specification is intended to address cars of the early 1980's. We did, however, allow for other options in photometrics, provided the maximum forward illumination does not exceed 170,000 beam candlepower at H-V. This appears to be a reasonable maximum for U.S. road conditions.

Vertical headlight-aim adjustment, dual intensity-signal lamps, and lamp-failure indicators are desirable options well within current technology. Other requirements are summarized in Figure 9. Specifically, we have prescribed the minimum number of controls and displays that we believe to be necessary for proper operation of the vehicle. The instrumentation is primarily required for test operations. However, we expect it to be integrated into the vehicle control panel design.

We have also required a One "g" omnidirectional restraint for the driver. This device is intended primarily to assist the driver in maintaining control during test maneuvers. However, we can see safety advantages in real traffic situations, such as, evasive maneuvers; preventing driver ejection should a door inadvertently open as a result of a collision or door latch failure; and keeping the driver in front of the controls and primary restraint system.

Requirements for a number of warning devices have been included in the specification. We have required an audible speed-warning device as a driver aid. We have also required a passive-restraint system condition-monitoring circuit to insure the system is operable. We have also encouraged incorporation of devices which will warn of impending mechanical failures or other unsafe conditions. However, based upon present knowledge, we do not feel these should be mandatory.

Occupant compartment environmental considerations include carbon monoxide levels, subjective comfort levels, and air conditioning requirements. The carbon monoxide level specified, 25 ppm, is based upon NASA requirements. The need for a standard was indicated in the October 1971, *MVP Plan*.

We have required air conditioning, because the prototype ESVs maintained passenger compartment integrity with fixed windows and the resulting lack of ventilation made these cars unsatisfactory.

Subjective comfort levels have been included because some of the prototype ESVs sacrificed ventilation, noise, and vibration insulators to save weight. We are attempting to obtain an automobile which is more representative of a marketable product.

A new section, designated emergency equipment, is self explanatory and will insure that a reasonable provision for such equipment exists. We consider this equipment to be included in the curb weight.

Turning now to the major category of Crash Energy Management, a number of important changes have been made (Figure 10).

There have been significant reductions in design conditions for frontal-collision modes. These reductions have been made to provide latitude for development of structural designs with reduced weight.

SAFETY
CRASH ENERGY MANAGEMENT

- DESIGN CRASH MODES
  - FRONT BARRIER 50 MPH
  - FRONT POLE 30 MPH
  - DOLLY ROLLOVER
  - REAR MOVING BARRIER 27.5 MPH
- PASSENGER COMPARTMENT INTRUSION LIMITS RELAXED WHEN CONTROLLED
- MAXIMUM ALLOWABLE PASSENGER COMPARTMENT ACCELERATIONS INCREASED AT LOW SPEEDS
- CAR TO CAR CRASH PERFORMANCE REQUIREMENTS DEFINED

Figure 10

Frontal rigid flat barrier impact requirements are now designated at the original 50 mph crash speed *only* in the case of perpendicular impacts. For oblique impacts, the design crash speed is reduced linearly with the angle varying from 50 mph at 0° to 30 mph at 45°.

Additional relief in frontal design conditions has been provided for fixed rigid pole design impact speed. This speed is now at 30 mph, a substantial reduction from the original 50 mph requirement. Special attention should, however, be given to the fact that pole impact has been made a design consideration extending over the *full* width of the vehicle, rather than applying only to the central half of the bumper, as had been the case in earlier specifications. Angular pole impact considerations have been eliminated from these new design specifications since it is well recognized that frontal design conditions exert the strongest influence on the final economy of the vehicle design. We expect and invite your comments on this portion of our specification.

Rollover design requirements are to be dictated by a 30 mph dolly-mounted vehicle rollover test.

The rear-end design requirements have been relaxed. The car-to-car, front-to-rear closing speed has been reduced from the original 75 mph to 50 mph. The 4,000 pound moving barrier closing speed has been reduced from the original value of 50 mph to 27½ mph at 0° and is reduced linearly as a function of angle to a value of 20 mph at 45°. Fixed-pole rear impacts have been deleted as a design condition in this specification.

Latitude has been provided to designers in the area of passenger compartment intrusions. Controlled-magnitude intrusions into the passenger compartment will be allowed when these intrusions are shown to create no additional hazard to the occupants.

Less constrictive design performance requirements have been designated for passenger compartment maximum accelerations as a function of fixed-rigid, barrier-impact speed (Figure 11). The new specification allows increased compartment accelerations at lower impact velocities. This will provide greater latitude to the vehicle designer in his selection of energy-absorption systems. However, it should be emphasized that this figure represents *maximum* design acceleration values and, therefore, implies the acceptability and desirability of lower compartment accelerations.

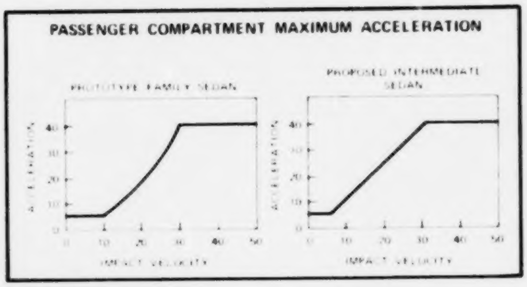


Figure 11

Car-to-car collision performance requirements are of paramount importance in light of the fact that accident statistics reflect a much higher frequency of occurrence for this type of collision than for collisions with fixed obstacles. We addressed this problem in our prototype family sedan specification but the performance requirements involved collisions only between ESVs of the same weight. We are now attacking this problem with consideration given to struck-vehicle weights ranging from 1,500 to 8,000 pounds.

Figure 12 reflects the essence of these requirements. Generally stated, our 3,000 pound ESV design must incorporate characteristics which preclude vehicle acceleration and crush distance excursions beyond specified limits in either car during car-to-car frontal collisions at the minimum closure velocities indicated.

Figure 13 provides a number of clarifying statements and assumptions that are necessary for a complete understanding of the graph. The majority of these statements are self-explanatory, but I would like to comment on two of them. First, we feel that the figure chosen for the ratio of average acceleration to maximum acceleration is a realistically attainable goal, based on our family sedan experience. Second, our selection of struck-vehicle designs is dictated by a limit of 30 g's for the maximum acceleration in 50 mph barrier collisions. This selection is based on two considerations: 1) a review of work done by a number of participants in this conference; 2) a belief that a 30 g limit will not impose excessive requirements on the occupant restraint system.

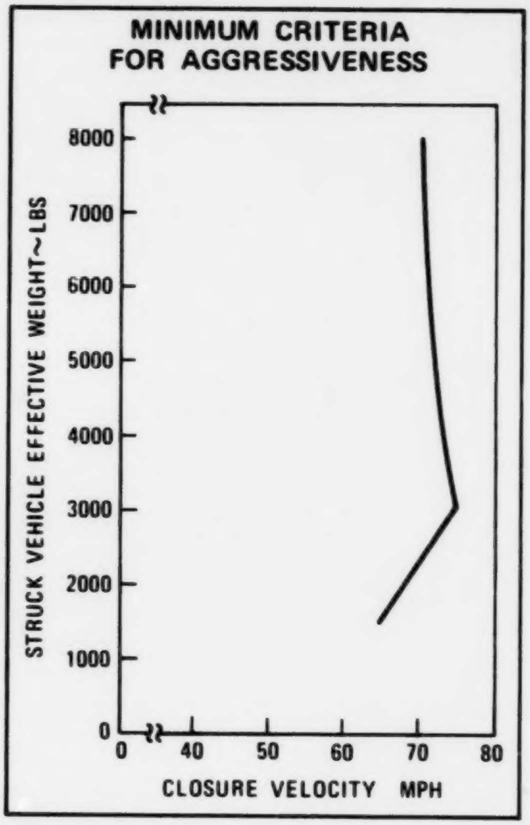


Figure 12

ESV DESIGN SATISFYING FIGURE VI

CLOSING VELOCITIES LIMIT MAXIMUM VEHICLE ACCELERATIONS AND CRUSH TO DESIGN VALUE

ASSUMPTIONS

- ESV EFFECTIVE WEIGHT = 3000 LBS
- AVERAGE ACCELERATION/MAX ACCELERATION = 0.85
- STRUCK VEHICLES DESIGNED FOR 30g MAX IN 50 MPH FIXED BARRIER COLLISION
- HEAD ON COLLISION AND ENERGY ABSORBING CRUSH EQUAL TO BARRIER IMPACT CRUSH
- STRUCK VEHICLES DESIGNED IN ACCORD WITH PRINCIPLES OF FIGURE VI

Figure 13

This represents our best attempt at specifying compatibility between different weight vehicles. Although the present specifications relate to collisions between advanced safety vehicles, it is anticipated that the format of these specifications will also assure reasonable compatibility between advanced safety vehicles and current conventional vehicles, during those critical years when the traffic mix will include both. Your special consideration and comments concerning this area are essential. The implications of these requirements on resulting accommodating vehicle designs are most significant.

That portion of the specification pertaining to the seating system is in compliance with existing standards (Figure 14). It also gives recognition of the potential advantages of novel design approaches, for which alternate acceptance criteria are provided.

SAFETY
OCCUPANT COMPARTMENT

- SEATING SYSTEMS REQUIREMENTS INCORPORATE AVAILABLE STANDARDS
- RESTRAINTS
  - FRONT REQUIRED TO BE PASSIVE
  - REAR MAY BE ACTIVE
- FLAMMABILITY, INTERIOR DESIGN INCORPORATE STANDARDS AS AVAILABLE

Figure 14

Occupant restraint requirements have been modified. The new specification reflects more realistic priorities for the various seating positions, based on frequency of occupancy and injury, as determined from available accident statistics.

The specification still requires passive-restraint systems for the driver and front seat occupants. Rear outboard passengers are assigned higher priority than a center rear passenger. Active restraint systems are recognized as potentially suitable for rear seat applications.

The flammability requirements primarily utilize currently existing safety standards.

Two additional crash-energy management performance requirements are pedestrian/cyclist protection and no damage impact velocities (Figure 15).

Pedestrian/cyclist protection requirements are presented in greater detail here than in the prototype specification. We are attempting to control external vehicle geometry, and we are requiring the incorporation of energy absorption which will limit pedestrian/cyclist acceleration levels in impacts up to 25 mph. Additionally, we have included goals for pedestrian/cyclist trajectory control during impacts above 10 mph. These requirements are based on a very limited amount of data, plus a number of assumptions. Therefore, your comments and recommendations are particularly solicited in this area.

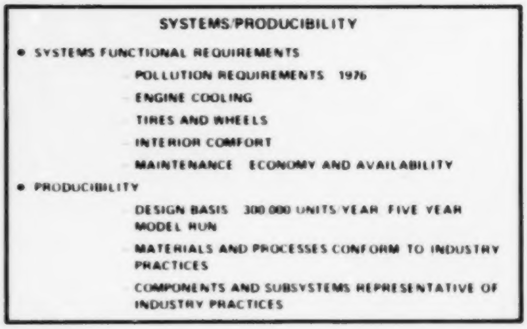
SAFETY
CRASH ENERGY MANAGEMENT

- PEDESTRIAN/CYCLIST PROTECTION REQUIREMENTS MORE EXACTLY SPECIFIED
- NO DAMAGE SPEED REDUCED TO 5 MPH

Figure 15

Frontal and rear no-damage impact speeds have been reduced from 10 mph to 5 mph.

The preceding charts have dealt with safety performance. While safety is the main concern it is, nevertheless, essential that we define a fully functional car (Figure 16). We want a car that satisfies the consumer and that complies with environmental standards. We have specified the 1976 exhaust emissions level.



SYSTEMS/PRODUCIBILITY

- SYSTEMS FUNCTIONAL REQUIREMENTS
  - POLLUTION REQUIREMENTS 1976
  - ENGINE COOLING
  - TIRES AND WHEELS
  - INTERIOR COMFORT
  - MAINTENANCE ECONOMY AND AVAILABILITY
- PRODUCIBILITY
  - DESIGN BASIS 300,000 UNITS/YEAR FIVE YEAR MODEL RUN
  - MATERIALS AND PROCESSES CONFORM TO INDUSTRY PRACTICES
  - COMPONENTS AND SUBSYSTEMS REPRESENTATIVE OF INDUSTRY PRACTICES

Figure 16

All vehicle functional systems must be properly designed to perform their intended functions. For example, on some of our prototype family sedan cars, the engine cooling was inadequate.

Interior passenger comfort is also important to the car occupant, and we have indicated, here, those items of comfort that are desirable courtesy lighting, glove compartment, radios, and clocks, along with control of noise, temperature and vibration.

We have also included in this section the basic requirement for ease and economy of maintenance. While this may not always be feasible in a research car, we nevertheless feel that consideration must be given to these problems, especially for such routinely replaced components as brake pads or linings, tires, batteries, anti-freeze, and lamps or bulbs. Costs of our prototype test program were kept down because both Goodyear and Firestone generously kept us supplied with tires.

We have attempted to address the problem of "producibility" in greater detail in this specification than in the prototype family sedan. A production basis of 300,000 units a year and a five-year production run were given as indicators of the level of manufacturing systems effort required. All materials and processes must conform to the expected industry practices, as well as the parts and components used. This does not mean that tooling must be capable of this production; only that the design be suitable for such manufacturing. Again, we welcome your suggestions in this area.

In conclusion, I would like to review some of our thoughts on using this specification.

First of all, we anticipate that the design/development will be performed by a contractor selected through competitive procurement. We foresee a greater level of government and industry participation in the design than was the case with the prototype family sedan. And, we do not intend to have two or more contractors in direct competition. There will be weight and performance incentives in the contract.

In the crucial area of weight, we foresee an allocation of weight and continuous monitoring of the status as the design progresses. A related benefit of this effort is early assessment of weight variations on vehicle dynamics through use of computer simulations.

In closing, I would like to reemphasize that the 3,000 pound ESV specifications are preliminary in nature at this stage of their development, and in fact, should be considered as research goals. They are goals which we believe will significantly advance the cause of safety and will influence the design of vehicles in the 1980's. We sincerely invite your participation with us in further developing the specifications for this class of experimental safety vehicle.

THE SWEDISH TECHNICAL PRESENTATION

SECTION TWO PART 6

INTRODUCTION

GUSTAV EKBERG, *Chief Vehicle Division*
National Traffic Safety Agency

The Swedish ESV Technical Presentations this afternoon contain five items. Mr. Rundkvist, who is the leader of the Swedish ESV Project, will give a report on statistical investigations carried out as the first step of the project "Steerability During Emergency Braking." The aim is to evaluate the benefits of the anti-lock braking system.

Mr. Jaksch, Project Manager of "Handling and Stability" at the Volvo Company, will speak of "Chassis Parameters Influence on Handling Characteristics Computer Simulations for the Volvo Experimental Safety Car," which we call VESC.

Mr. Saxmark, Project Manager of Interior Safety at Volvo, will present a report on the "Airbag Development for the Volvo Experimental Safety Car."

A report on "Statistical Traffic Accident Analysis with Reference to Occupant Restraint Values and Crashworthiness" of the Volvo Experimental Safety Car will be presented by Mr. Asberg, Project Manager, as the author, Mr. Bolin was unable to attend this conference.

Finally, Mr. Gustavsson, Technical Director at the Saab-Scania Company, will speak on Saab-Scania "Road Accident Investigation."

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THE SWEDISH TECHNICAL PRESENTATION

SECTION **TWO** PART **6**

STEERABILITY DURING EMERGENCY BRAKING

STEN RUNDKVIST, *ESV Project Leader*
AB Volvo

The Swedish ESV program is an investigation of "Steerability During Emergency Braking" and is divided into the following task groups:

- Accident investigation
- Simulation testing by a mathematical model
- Field testing performance tests
- Field testing statistical tests.

The objective of the program is to prove the safety benefits of steerability during emergency braking and, if any, to suggest performance standards and recommend test procedures.

ACCIDENT INVESTIGATION

The objective of the accident investigation task was to find the percentage of locked-wheel accidents; to find typical situations for locked-wheel accidents; and to estimate the possibility of preventing or reducing the consequences of the accident, if steerability during braking had been possible.

This investigation has been completed and three reports are available: *Study of Police Reports, Interviews* and *Summary Report*. The last report also includes a "literature study."

All police reports on motor vehicle accidents during 1971, in a given police district, have been studied and all accidents where locked wheels were reported have been data processed. The police district can be regarded as typical for southern Sweden, as it includes towns and villages as well as countryside. The roads are of different types and in various conditions.

The interviews were primarily aimed at clarifying types of accidents, or incidents involving brake application with locked wheels, and were held mainly with drivers of company cars at Saab-Scania and Volvo. A trial investigation by mail among statistically selected people gave such a discouraging result that the idea was abandoned.

Those who took part in the interview were asked by letter whether they, as car drivers, had ever applied the brakes in such a way that they found themselves in a situation which they considered critical, or in which they lost control of the vehicle. Those who gave a positive answer were then interviewed through a special questionnaire. The accidents/incidents were subdivided into categories and coded before being processed by computers.

For the literature study, a search and analysis was carried out. The search was largely performed with computer aid. About 50 references were received and analysed. Furthermore, personal contacts were made with institutions and companies working with accident analysis. Only three reports were of value to our investigation.

ACCIDENT INVESTIGATION

The first objective of this task group was to find the proportion of accidents involving locked wheels.

The police reports showed that 10.5 percent of all reported accidents had locked wheels. As this study is based only on the information in the police reports, there probably were accidents with locked wheels which were not indicated in the reports.

The literature study gave 15 percent as an average of locked-wheel accidents.

Conclusion: The lower limit for locked-wheel accidents is 10 percent with a probable range of up to 15 percent of total number of accidents.

The second objective was to find typical situations for locked-wheel accidents. According to the police report, of the 168 accidents investigated, the most common situations were "Collision with vehicle in road junction," 33.3 percent, and "Running into vehicle from behind," 19.0 percent.

Of the 372 cases investigated by interviews, the most common situations were "Running off road due to high speed," 22.8 percent; "Running into vehicle from behind," 22.3 percent; and "Collision with vehicle in road junction," 11.3 percent. The interviews, however, included incidents with no damage. They amounted to 51.9 percent, and it is interesting to note the distribution among the three accident situations, when only accidents with property damage or personal injury were involved.

Among the remaining 179 accidents there were the following percentages: first, "Running into vehicle from behind," 25.7 percent (earlier second place);

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second, "Collision with vehicle in road junction," 18.4 percent (earlier third place); third, "Running off road due to high speed," 12.8 percent (earlier the most common accident type).

According to the interview, "Running off the road due to high speed" involved the highest percentage of accident situations with 22.8 percent, while real accidents involved only 12.8 percent. The police reports give no percentage figure, but it must be rather small. Those accidents were mostly single-car accidents and the driver certainly was not interested in getting the police involved, which may explain the difference.

The most frequent accident situations, "Collision in road junction" and "Running into vehicle from behind" were used in the statistical tests, as will be seen later.

Typical road conditions for locked-wheel accidents were:

- Ice or snow: police report, 38.7 percent, and interviews, 52.4 percent. The rest were divided almost equally between dry and wet road.
- Straight road: police report, 72 percent, and interviews, 59.1 percent. Road junction is classified as straight road.
- Estimated speed up to 50 km/h (30 mph): police reports, 57.7 percent, interviews, 44.4 percent.

The driving license was three years old or less in 32.1 percent of the locked-wheel accidents, according to the police reports.

Conclusion: The typical locked-wheel accident happens to a vehicle driven by a new driver, who approaches a road junction at around 50 km/h (30 mph) on a slippery surface and meets another vehicle from the right or left.

The third objective was to find the use of steerability during locked-wheel accidents.

First, I want to define the word "prevention." In the police reports, "prevention" is considered the ***investigator***'s judgment as to whether the accident would have been avoided, had the vehicle been steerable; in the interviews, it is considered the ***driver***'s judgment. In both investigations it was 70 percent.

The driver attempted to steer in 50 percent of the cases, according to the police report and in almost 70 percent of the cases in the interviews. From this basic data it is possible to determine in how many cases the driver attempted to steer when the prevention was good. The result is 40 percent in the police report and almost 50 percent in the interviews. If, then, 40 to 50 percent of the locked-wheel accidents could have been avoided and 10 to 15 percent of all accidents were locked-wheel accidents, it can be concluded that from four to 7.5 percent of all traffic accidents could be avoided or considerably reduced if the vehicles were steerable during emergency braking.

SIMULATION TESTING BY A MATHEMATICAL MODEL

This task group had as its objective: to complement the field tests and to perform a systematical investigation of the influence of typical anti-skid parameters.

The vehicle model was simulated in the hybrid simulator in which Volvo and Saab cars are simulated. However, the model is formed in such a way that different types of vehicles can be studied by variation of parameters.

Three anti-locking systems, representing different manufacturers, can be programmed in the digital part of the simulator and it is easy to vary system parameters such as:

- Individual regulation
- Select low on the rear wheels
- Select low on all four wheels

It is also easy to vary different specific anti-locking parameters to find the influence on the vehicle behavior. Examples of parameters:

- Switching criteria (wheel deceleration, acceleration and velocity level)
- Switching time delay
- Hydraulic rise and fall time

The functions describing the forces between tire and surface are stored in the digital part of the simulator. The forces are u-curves, representing braking force as well as side force, measured under variations of the following parameters:

- Surface (ice, dry asphalt, skid pad)
- Tire on ice (summer tire, studded tire)
- Velocity
- Slip angle
- Normal force

The model is now under validation against available results from the field tests on high u.

The u-curves used in the simulator are partly from our own measurements of braking force and partly from curves presented in the literature for side force. There are, however, no curves available representing the forces on ice and, since there was no measuring equipment available, we had to complement our own equipment. The intention was to measure u-curve at the same time and place where the performance tests were going on, to get as good a correlation as possible between field tests and simulations. The measurements will start on ice at the end of March, this year.

FIELD TESTING – PERFORMANCE TESTS

The third task group, concerned with performance tests, systematically investigated the relationship between braking (longitudinal deceleration) and steering capability (lateral acceleration), under different conditions.

The variables used in the test were:

- Vehicle type: front wheel driven, engine in the front; rear wheel driven, engine in the front; rear wheel driven, engine in the rear.
- Anti-skid device: from three different manufacturers.
- Friction coefficient: very low u (ice and summer tires); low u (ice and studded tires); high u (dry asphalt); variable u (skid pad).
- Velocity: ice, 50 and 70 km/h (30, 45 mph); dry asphalt, 70, 90 and 110 km/h (45, 55, 70 mph); skid pad, 50 and 70 km/h (30, 45 mph).
- Load condition: driver with measuring equipment; maximum load.

The test procedure for each configuration was:

- Braking straight ahead without anti-skid device, which meant locked wheels.
- Braking straight ahead with anti-skid device.
- Braking in a curve, with the steering wheel locked at a position giving minimum free-rolling radius. The brakes were applied about one second after turning the steering wheel.

During the braking in a curve, an anti-locking device was necessary; otherwise, the vehicle would have continued straight ahead in the tangential direction. This test procedure was chosen because it was well defined, reproducible and independent of the driver's skill. Each individual drive was repeated three times.

The measuring equipment consisted of a platform stabilized in roll and pitch measuring. Longitudinal deceleration, lateral acceleration, yaw velocity and the measured values were tape recorded for future data processing.

Some trial tests were performed on high u in the fall to check the test procedure and data program. The systematical testing started in January of this year with tests on ice.

Some diagrams made from tests on high u while braking on a curve may be of interest. Figure 1 shows longitudinal deceleration as a function of time, for two different types of vehicles under the same condition, with the exception that vehicle B has a five percent smaller minimum free-rolling radius than vehicle A.

When brakes were applied on dry asphalt, the velocity was 109 km/h (70 mph) and almost reached 1 g at the end of the braking.

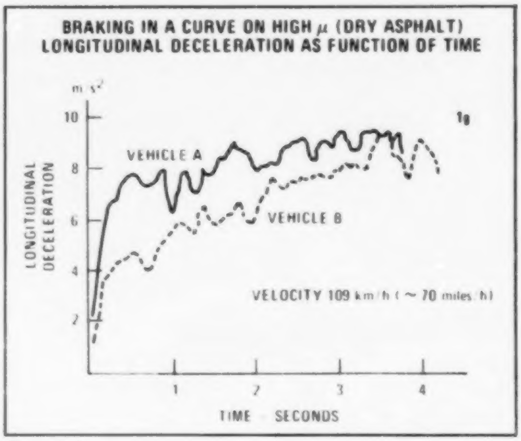

Figure 1

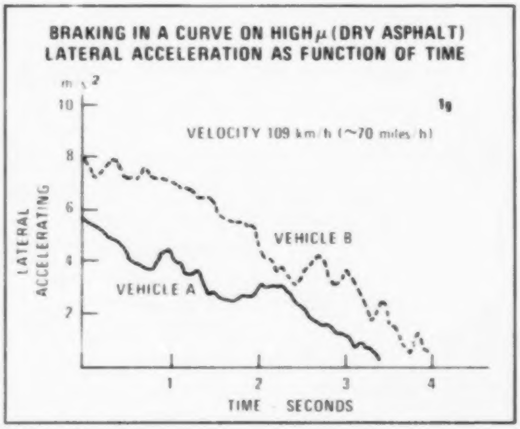

Figure 2

Vehicle B had a lower deceleration at the beginning but that was because a greater part of the available friction was used for lateral acceleration (side force, steering capability) which is shown in Figure 2.

The vehicle track, as calculated from the registered data on the tape, was plotted on an X-Y diagram, together with the arc of the circle with minimum free-rolling radius (dotted line). The X-axis was in the velocity direction at the start of braking.

Figure 3 shows that the braking distance, measured from the time when the brakes were fully applied until the vehicle has stopped, was for vehicle A 59.7 m (197 ft.); the lateral deviation was 11.5 m (38 ft.).

The braking distance for vehicle B (Figure 4) was 67.8 m (222 ft.) and the lateral deviation 21 m (69 ft.).

The four diagrams together give a better survey.

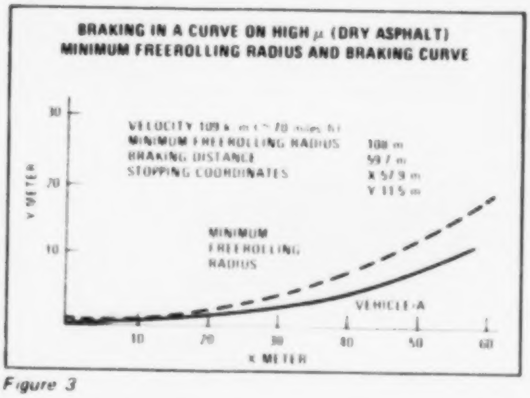

Figure 3

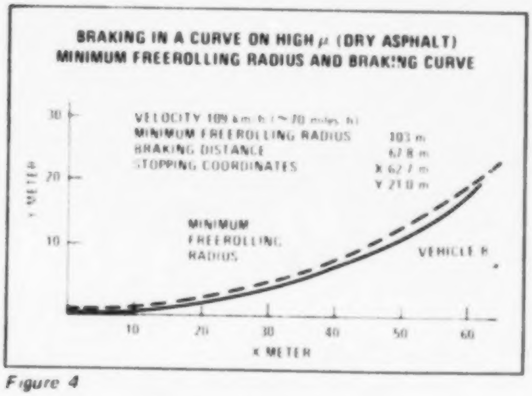

Figure 4

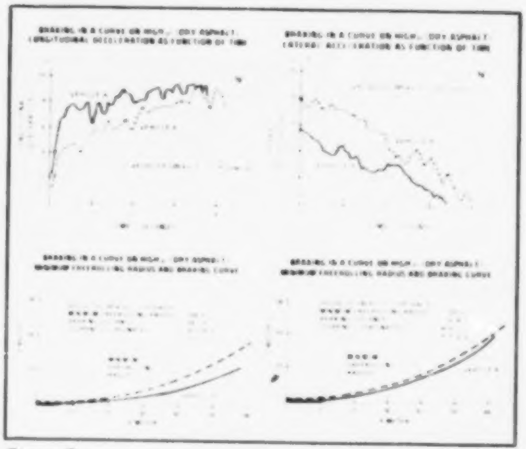

Figure 5

BRAKING ON HIGH (DRY ASPHALT)
VELOCITY 109 km/h (~ 70 miles/h)

	STOPPING DISTANCE	AVERAGE RETARDATION
BRAKING STRAIGHT AHEAD LOCKED WHEELS	64.1 m	7.15 m/s^2
BRAKING STRAIGHT AHEAD ANTILOCKING SYSTEM	55.9 m	8.20 m/s^2
BRAKING IN CURVE ANTILOCKING SYSTEM	59.7 m	7.66 m/s^2

Figure 6

It is evident from Figure 6 that if braking is more accentuated there will be less force available for side deviation, and vice versa. The braking distance was measured for vehicle A under the same conditions as for braking straight ahead with and without an anti-locking device and for braking on a curve.

The braking distance on a curve 59.7 m (196 ft.) lies between braking straight ahead with locked wheels 64.1 m (210 ft.) and braking straight ahead with an anti-locking system 55.9 m (183 ft.). These are only a few examples from the trial runs. The systematical evaluation has not yet started.

FIELD TESTING – STATISTICAL TESTS

The task group on statistical tests sought to determine how an average driver behaves during an unexpected emergency situation and whether or not having steerability works to his advantage. .

The emergency situation was established through the use of a plastic foam dummy car, which was concealed from the test driver's view. When the test car reached a certain point on the road a device was activated which pulled the dummy car in front of the test vehicle, at a distance that would permit a very skilled driver to avoid the obstacle even without anti-locking brakes. This situation simulated the most common accident situations found by the accident investigation group: "Collision at a road junction," and "Running into a vehicle from behind."

The driver was given the impression that the test would be concerned with his reaction to traffic lights and speed signs, while driving at 70 km/h (45 mph). This provided the same conditions for the different drivers.

The driver first performed some ordinary driving to get acquainted with the car before entering the special track where the traffic lights and speed signs were located. The track was on a frozen lake with snow walls around, so that there was no danger of accidents. An instructor accompanied the driver in the vehicle during the whole test. The driver was unaware that the car had an anti-locking system.

During the first lap, the dummy car suddenly appeared and, during the braking period, the following quantities were registered or measured:

- Steering wheel movement

- Indication of when braking pedal was depressed
- Wheel speed when anti-locking system was functioning
- Reaction time and distance measured from the time of activating the dummy car to application of the brakes
- Final stop position

The driver received a short lesson about the functioning of anti-locking systems and then drove a few more laps, during which the dummy irregularly appeared again. When he became accustomed to the anti-locking system, it was disconnected without his knowledge. When the dummy car appeared, the brakes did not react as he had become accustomed to, and still another type of unexpected situation was created.

The tests are still going on and the intention is to test at least 35 drivers. There has been no time to evaluate the results so far, but there are all kinds of reactions, from full brake going straight ahead to almost no braking but steering to the left or right.

The driver was also asked what his first reaction to the dummy was and there were varying impressions: child, concrete block, reindeer, small car, railway wagon. So I think we really have succeeded in creating an unexpected situation.

THE SWEDISH TECHNICAL PRESENTATION

SECTION TWO PART 6

CHASSIS PARAMETER INFLUENCE ON HANDLING CHARACTERISTICS. COMPUTER SIMULATIONS FOR THE VOLVO EXPERIMENTAL SAFETY CAR.

FRIEDRICH JAKSCH, *Project Manager*
Handling and Stability
AB Volvo

INTRODUCTION

The VESC project, which started at Volvo in 1969, is a tool to do research on a great number of vehicle safety aspects. One part of the planned work in the VESC project is to analyze and optimize handling properties for a vehicle of Volvo's size. This handling work includes, among others, measuring vehicle behavior through subjective judgment as well as computer simulations.

This report is a computer simulation and will indicate the influence of different vehicle parameters on handling and stability. The simulations are based as far as possible, on the requirements in the ESV specifications. Furthermore, the basic concept and the different chassis systems for the VESC, for example suspension and steering systems, are described.

The mathematical model which we are using here has five degrees of freedom and is, therefore, relatively simple. Some of the input data are non-linear functions, for example, the characteristics of tires, wheel suspension systems, shock absorbers, springs, etc. These parameters are measured on vehicles or on vehicle components. The correlation between computer tests and practical driving tests is fairly good and is indicated in the following diagrams. Figure 1 shows the steady state yaw response for the Volvo 140 production vehicle. As shown in the diagram, the test was carried out on a test track with a constant radius. Validation of the computer model for transient response conditions is seen in Figures 2 and 3, which show a lane change test on Volvo 140. The dimensions of the lane appear from the figure. During the test, the speed was a constant 80 km/h. In this lane change test we have used a mathematical model where the vehicle follows a given theoretical path, i.e. a "closed loop" system.

As can be seen from the diagrams, the correlation is good for stationary tests as well as for dynamic tests. This makes the mathematical model useful for simulations. A detailed description of the computer model is contained in the Appendix.

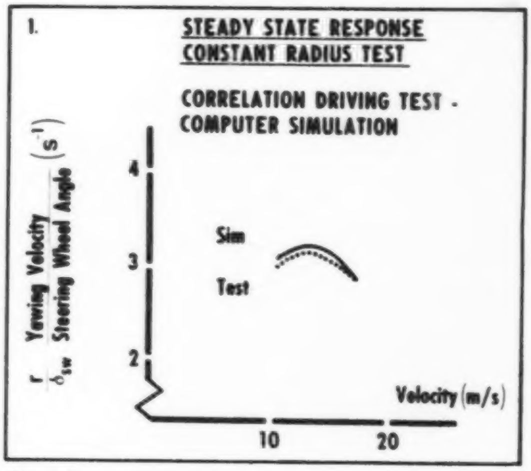

Figure 1

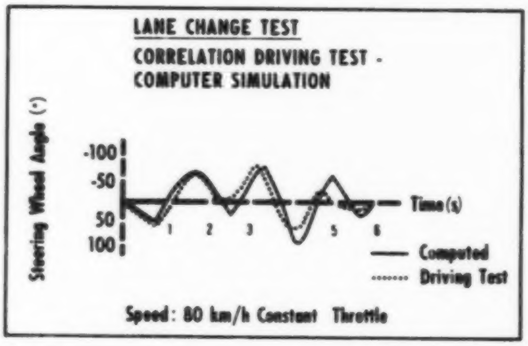

Figure 2

The parameter variations which we have used in this investigation, for example, caster offset, weight distribution, stiffness in the steering system, etc., are not larger than the designed variation in different makes of production vehicles.

COMPUTER SIMULATION OF VESC PROPERTIES AND A COMPARISON WITH ESV REQUIREMENTS

Steady State Yaw Response

The ESV requirements do not take into considera-

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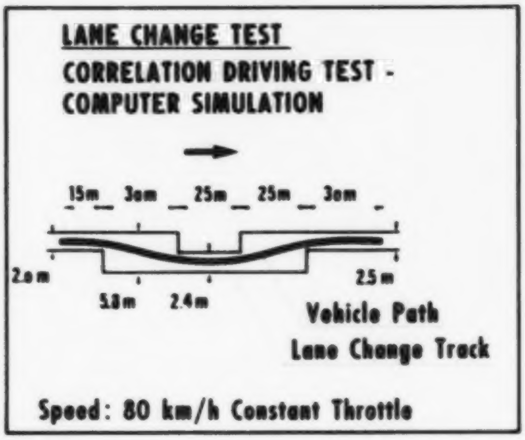

LANE CHANGE TEST
CORRELATION DRIVING TEST - COMPUTER SIMULATION
15m
3om
25m
25m
3om
2.o m
2.5 m
5.8m
2.4m
Vehicle Path
Lane Change Track
Speed: 80 km/h Constant Throttle

Figure 3

tion the relationship of steering wheel to front wheels.

Based on our experience, we maintain that the steering system cannot be neglected. The stiffness between the steering wheel and the road wheels is of great significance for the driver when judging the driving properties of a vehicle. A steering input at the front wheels, therefore, is not suitable.

Consequently, we specify the yaw response in relation to the steering wheel angle, with a steering gear ratio of i = 1.

This, of course, implies that it is not possible to compare the simulated results directly with the ESV requirements.

Figure 4 shows three different response curves for VESC: C = Manual steering and tires of type 175 HR 14; B = Manual steering and tires of type 185/70 HR 14; A = Power assisted steering and tires of type 185/70 HR 14.

As can be seen, the difference between these three alternatives is rather small. In practice, however, the difference between cars with power-assisted steering and manual steering may be much greater than demonstrated in this diagram. The reason for this is probably the difference felt in steering wheel torque between power assisted and manual steering. With respect to characteristics, the normalized yaw response at different lateral accelerations is of importance. Therefore, different lateral acceleration curves for the basic VESC are illustrated in Figure 5.

Transient Yaw Response

Also in this case, we specify the yaw response in relation to the steering wheel angle with a steering gear ratio of i = 1.

Figure 6 shows the transient yaw response with power assisted steering and tires of type 185/70 HR 14.

The stiffness between the steering wheel and the front wheels differs between power-assisted and manual steering systems. This difference affects the response of the vehicle. This can be felt especially in a car on which the spring constant is too low.

Figure 7 shows the transient yaw response of VESC with different stiffnesses between the steering wheel and the road wheels. These variable stiffnesses, as shown in Figure 7, are easily obtained by choosing different types of steering gears.

The yaw response for the lowest speed, i.e. at 40 km/h, is not shown here, as the response curve for that speed is not influenced when compared with VESC in standard version.

Another factor affecting the yaw response curve is the roll steer coefficient. Figure 8 shows the yaw

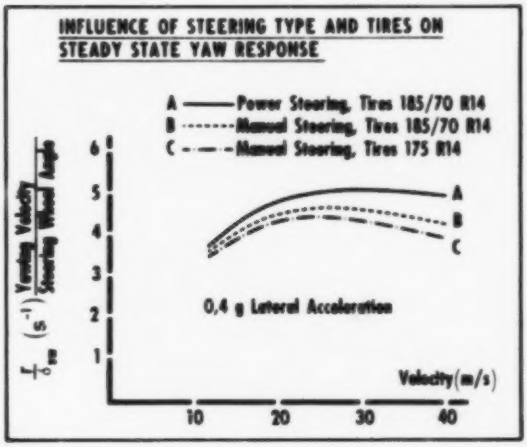


Figure 4

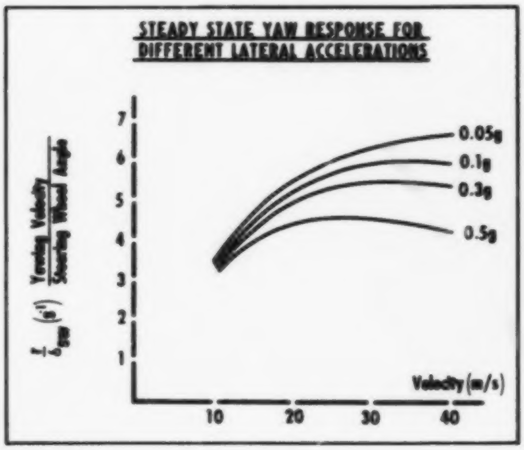


Figure 5

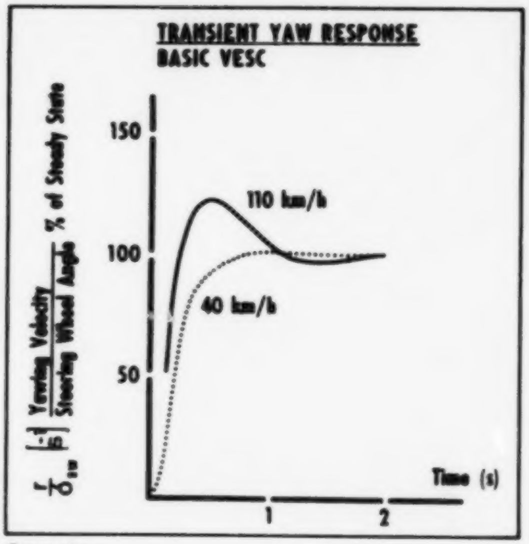


Figure 6

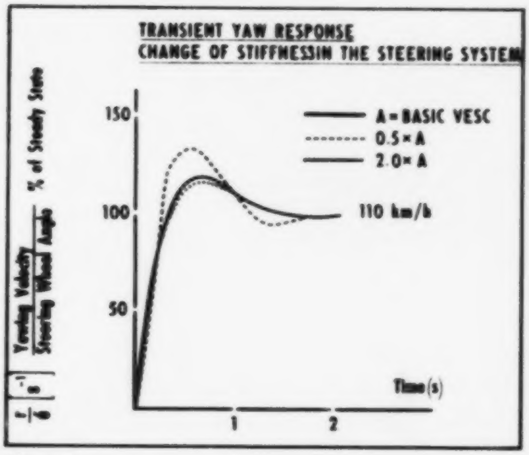


Figure 7

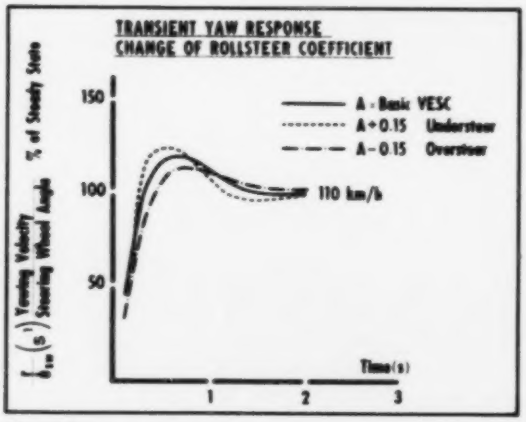


Figure 8

response variations with different roll steer coefficients. When driving on old country roads (low frequency disturbance), which still exist in some parts of Sweden, the driver's work in turning the steering wheel is considerably increased. The driver activates the steering wheel, because he is able to feel the movement of the vehicle. These movements apparently arise with such a big offset in phase that the correction of steering does not give the desired efficiency.

The front wheel preset angles also affect the driving characteristics. As far as the yaw response is concerned, the caster angle, may cause noticeable changes.

Figure 9 shows how the yaw response is changed, when the geometric caster offset on VESC is altered by +11 mm and -11 mm. Furthermore, the tires produce a caster effect, which may vary from one make of tire to another within the same size.

Figure 10 shows how the change in characteristics of the shock absorbers affects the yaw response. A reduction by approximately 50% of the roll-damping characteristics does not alter the yaw response, when compared with the standard version of VESC. Characteristics four times as big would cause a somewhat higher understeer.

Within the actual roll-damping area, there is a lower limit below which the vehicle would behave in a less satisfactory manner.

Upwards there is no definite limit, from the handling point of view, but the ride might be somewhat less comfortable. In this connection, we would like to point out that one may also feel a

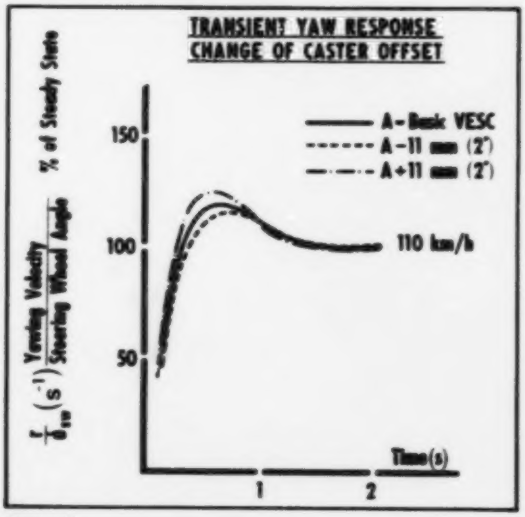


Figure 9

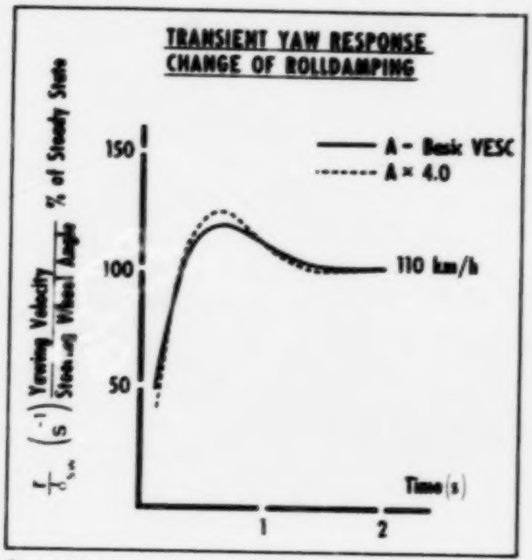


Figure 10

difference in roll characteristics, which is not present in the yaw response.

Figure 11 shows how the yaw characteristics of VESC are changed, if the roll stiffness is reduced by 50%. A low roll stiffness not only affects the driving ability in a negative direction, but also impairs the properties as far as overturning is concerned. A low roll stiffness results in a high remaining kinetic energy about the roll axis, when the bump rubbers get in contact with the axles. This may cause the vehicle to overturn.

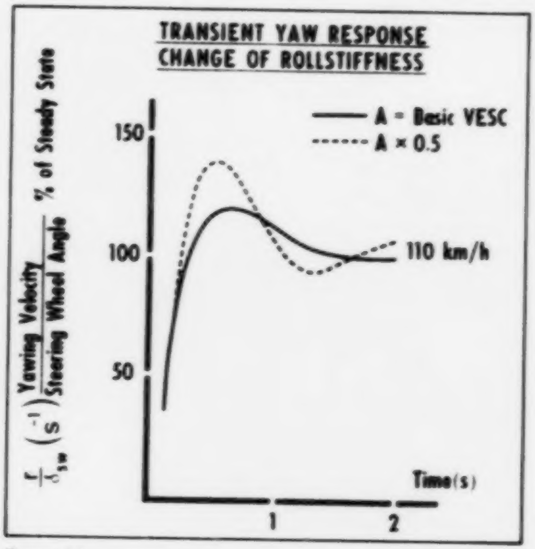


Figure 11

Figure 12 shows the yaw response for VESC with different loads. The diagram demonstrates the reduction in understeer for the full load case.

Figure 13 indicates the consequences which a faulty mass distribution of the car may have. This could happen if, for instance, too much weight has been put into the luggage compartment.

From Figures 12 and 13 it may be noted that special attention should be paid to the complexity of load distribution, partly because the change in yaw response is quite high and partly because the driver is personally able to control the load distribution. In the ESV specifications, only one load is used, which is normal weight, when measuring the handling characteristics.

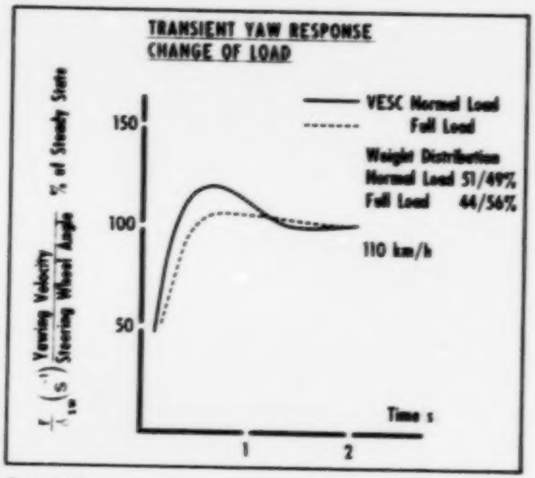


Figure 12

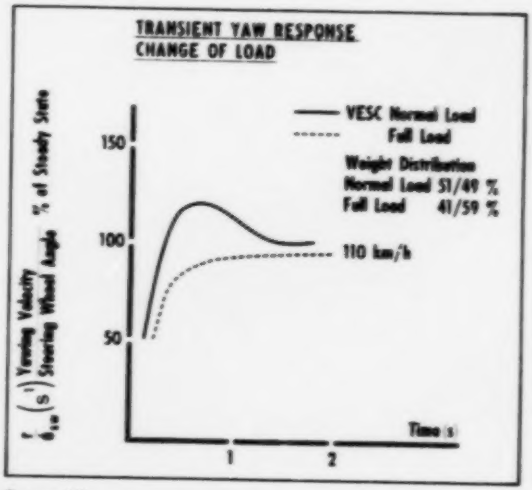


Figure 13

It is therefore recommended that the characteristics should be evaluated even at full load, since the difference in driving the same vehicle at normal load and full load may be greater than the difference between two completely different cars in their design load positions.

Crosswind Sensitivity

Figure 14 illustrates the course deviation with locked steering wheel for a number of VESC versions. The lowest course deviation has been obtained with tires of size 185/70 R 14 and power-assisted steering. With manual steering and tires of size 175 R 14 the course deviation is increased by approximately 22%.

The reason for this difference in deviation might well be the fact that cars with power-assisted steering have a higher stiffness between the steering wheel and the road wheel as compared with cars equipped with manual steering. Furthermore, tires of size 185/70 R 14 have a higher cornering stiffness than tires of size 175 R 14.

The diagram demonstrates that a fully-loaded vehicle has less deviation than a normally loaded one. This is due to the fact that the first inertia-opposed lateral acceleration is higher for a fully-loaded vehicle than for a normally-loaded one. This results in a quicker road wheel angle return of the fully-loaded vehicle, and thus less deviation occurs. In spite of the lower course deviation with a fully-loaded vehicle, one does not feel it to be more stable when driving. A possible reason for this is that a fully-loaded vehicle also makes other kinds of movements, for example roll and yaw, which result in a less satisfactory feeling.

Figure 15 shows the influence of different caster offsets on course deviation. As may be seen in the diagram, a negative caster offset gives less course deviation at 50 km/h and 80 km/h and approximately the same deviation at 110 km/h, when compared with a positive caster offset of the same magnitude.

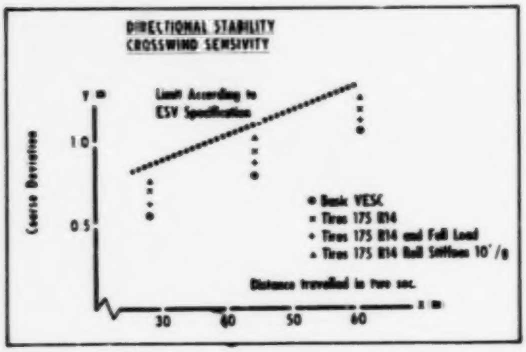


Figure 14

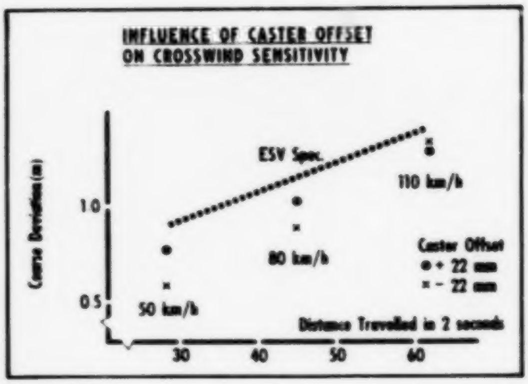


Figure 15

If we introduce a controlled feedback system in the mathematical model, and try to follow a straight line, when the vehicle is disturbed for 0.2 seconds by a wind gust according to the ESV specifications, we will get results as shown in Figures 16 and 17. A negative caster offset gives a course deviation of about 0.5 m maximum, instead of 0.6 m for the vehicle with positive caster offset.

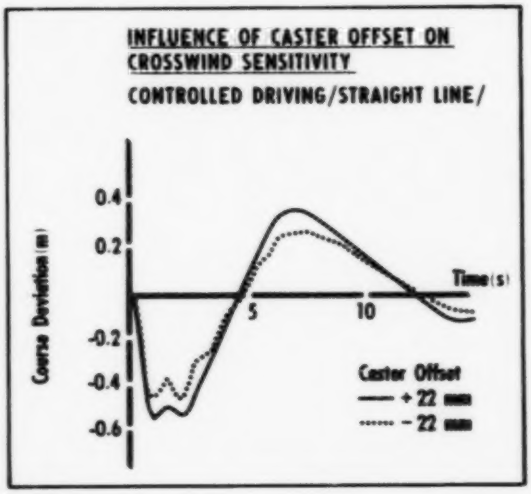


Figure 16

The steering wheel angle in Figure 17, for the vehicle with negative caster offset, is much more undamped, which indicates a lower yaw damping and therefore, more activity for the driver.

A vehicle with negative caster offset gives less course deviation under ESV test conditions than a car with positive caster offset, but the driver feels a vehicle with negative offset as more unstable. Our experience from practical tests confirm these results from simulations.

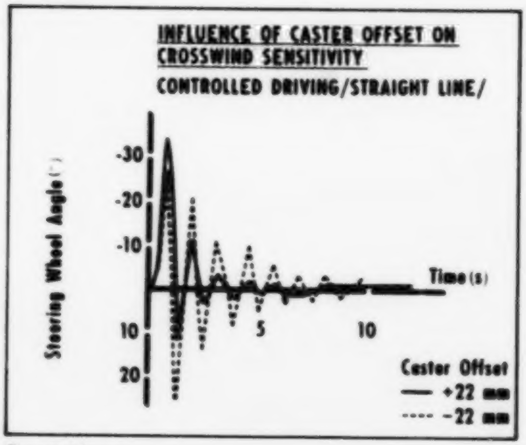


Figure 17

SUMMARY AND CONCLUSIONS

Changes of chassis parameters, as for example stiffness in the steering system, rollsteer coefficient, caster angle, tire characteristics, etc., give relatively small or moderate changes in the handling properties, as compared with the ESV requirements. Load changes, especially a change of the weight distribution, lead to relatively large changes of the handling properties.

From our point of view, the ESV requirements should, therefore, take the fully-loaded condition into consideration (Figure 18). This is of great importance because the loading conditions can be affected by the driver. A change in loading condition should give as small as possible changes in handling properties.

The crosswind sensitivity requirement, according to the ESV specification, can, in some cases, lead to bad correlation between subjective judgment and measurements. A vehicle with good crosswind sensitivity should have small course deviation but also high yaw damping or low steering wheel activity. The ESV requirements for crosswind sensitivity should be complemented with a requirement which, for example, would keep the yawing velocity within specified limits, in addition to the course deviation limit.

THE ESV REQUIREMENTS SHOULD TAKE THE

FULLY LOADED CONDITION

INTO CONSIDERATION

Figure 18

The basic VESC concept, with which we made comparisons in this investigation, has the following fundamental features (Figure 19):

VESC FUNDAMENTAL FEATURES

1. **COMPROMISE BETWEEN TRACTION AND STABILITY**
2. **TOTAL ROLLSTEER COEFFICIENT - ZERO AT NORMAL LOAD**
3. **FRONT ROLLSTEER COEFFICIENT - LOAD INDEPENDENT**
4. **INCREASED LOAD - INCREASED ROLL UNDERSTEER**
5. **HIGH STIFFNES - STEERING SYSTEM**
6. **LOW CAMBER CHANGE - FRONT WHEEL SUSPENSION**

Figure 19

- A weight distribution which gives good compromise between traction and stability under different loading conditions for a vehicle with rear wheel drive.
- A total rollsteer coefficient which is zero in the normal loading position, in order to give good directional stability on old country roads.
- A rollsteer coefficient on the front axle which is independent of load variations.
- A rollsteer characteristic of the rear axle, which gives increased roll understeer when the load is increased. A compromise has been made between weight distribution and rear axle rollsteer characteristic, which leads to fairly constant handling properties at different loading conditions.
- Relatively high stiffness between the steering wheel and the front wheels which provides exact steering and good directional stability.
- Low camber change in the front wheel suspension, due to body roll for the purpose of achieving a

Figure 20

good balance in slip characteristics together with the beam rear axle.

The different chassis systems which have been chosen for the VESC to obtain the above mentioned properties are as follows:

The front wheel suspension is a double wishbone system with the front wheel springs fitted on the wheel stub axle (Figure 20). In the rear, a beam axle has been chosen, suspended by means of five link rods. One of these links is fitted across the vehicle, as a Panhard-rod, to transfer lateral forces. The other four links are attached in the longitudinal direction of the vehicle to transfer tractive and braking forces (Figure 21).

The steering is of rack and pinion type with power-assistance (Figure 22).

The tires, which are of radial ply type, have the dimension 185/70 HR 14 and are fitted on wheels with a size of 5.5 J x 14.

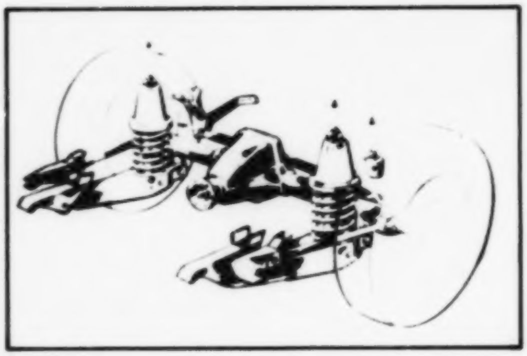


Figure 21

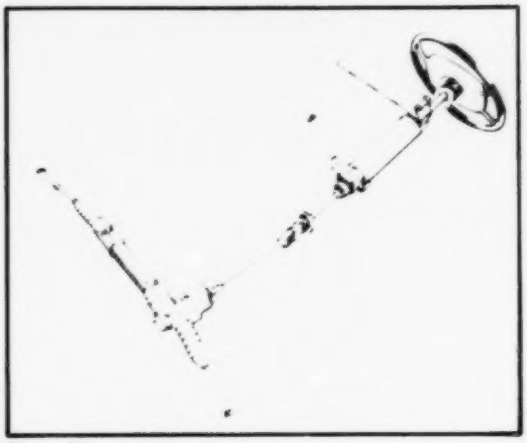


Figure 22

APPENDIX

1 SIMULATION MODELS

1.1 Basic Model

1.1.1 *Mathematical Equations*

x-axis $$\dot{u} = r \cdot v + \frac{1}{m}(X_A + X_R + X_T + X_B)$$

y-axis $$\dot{v} + \frac{m_s \cdot e}{m} \cdot \dot{p} = -r \cdot u + \frac{1}{m}(Y_A + Y_1 + Y_2)$$

Roll $$\dot{p} + \frac{I_{\varphi z}}{I_\varphi} \cdot \dot{r} + \frac{m_s \cdot e}{I_\varphi} \cdot \dot{v} = -\frac{m_s \cdot e}{I_\varphi} \cdot r \cdot u + \frac{1}{I_\varphi}(L_A + L_S + L_{SA} + L_B)$$

Yaw $$\dot{r} + \frac{I_{\varphi z}}{I_{zz}} \cdot \dot{p} = \frac{1}{I_{zz}}(N_A + N_{y1} + N_{y2} + N_{AT1} + N_{AT2} + N_{WT1} + N_{WT2})$$

1.1.2 *Designation*

$\dot{u}$ = Acceleration in direction of x-axis (in our model $=0$)
$\dot{v}$ = Acceleration in direction of *y-axis*
$\dot{p}$ = Roll acceleration
$\dot{r}$ = Yaw acceleration
u = Velocity in direction of x-axis (constant)
v = Velocity in direction of y-axis

p = Roll velocity
r = Yaw velocity
m = Total vehicle mass
m_s = Vehicle sprung mass
I_ϕ = Moment of inertia about roll axis
I_{zz} = Moment of inertia about z-axis
$I_{\phi z}$ = Product of inertia about roll- and z-axis
e = Minimum distance, center of gravity sprung mass — roll axis
X_A = Aerodynamic forces
X_R = Rolling resistance
X_T = Traction
X_B = Braking force
Y_A = Aerodynamic forces
Y_1 = Side force on front wheels
Y_2 = Side force on rear wheels
L_A = Aerodynamic moment
L_S = Moment of the springs
L_{SA} = Moment of the shock absorbers
L_g = Moment of the gravity
N_A = Aerodynamic moment
N_y = Moment caused by lateral force
N_{AT} = Self aligning torque
N_{WT} = Moment caused by an increased rolling resistance due to transfered loads.

1.1.3 *Input data*

1.1.3.1 Constant values

m = Total vehicle mass (kg)
m_s = Vehicle sprung mass (kg)
I_{zz} = Moment of inertia about z-axis (kgm^2)
I_ϕ = Moment of inertia about roll axis (kgm^2)
$I_{\phi z}$ = Product of inertia about roll axis and z-axis (kgm^2)
I_w = Moment of inertia, front wheels about their z-axis (kgm^2)
K_w = Spring constant, steering wheel — front wheels (at the wheels) (Nm/rad)
C_w = Damping constant, body — wheel (Nms/rad)
λ = Inclination of roll axis (rad)
l = Wheel base (m)
t = Track (m)
a = Distance, center of gravity for total mass to front axle (m)
e = Distance, center of gravity for sprung mass to roll axis (m)
h_f = Height of roll center, front (m)
h_r = Height of roll center, rear (m)
ε_1 = Roll steer coefficient, front (rad/rad)
ε_2 = Roll steer coefficient, rear (rad/rad)
$\partial X/\partial Z$ = Rolling resistance/vertical load (N/N)
u = Vehicle speed in x-direction (m/s)
ρ = Air density (kg/m^3)
A = Frontal area (m^2)
$\partial C_y/\partial\beta$ = Aerodynamic side coefficient (1/degree)
$\partial C_L/\partial\beta$ = Aerodynamic roll moment coefficient (1/degree)

$\partial C_N/\partial\beta$ = Aerodynamic yawing moment coefficient (1/degree)
C_x = Aerodynamic drag coefficient (—)
V_w = Side wind velocity
Δ_w = Max turning of steering wheel (degrees)
XM = Time for max turning of steering wheel (s)
$\left.\begin{matrix} N_1 \\ N_2 \\ M_y \end{matrix}\right\}$ = Coefficients for calculating the side force reduction due to the tractive power (—)
Z_R = Vertical load on one rear wheel (N)
T_{sw} = Time during which the sidewind is applied (s)
CAS = Caster offset (m)

When driving along a certain path the following is added:
I_1 = Moment of inertia of the steering wheel (kgm²)
I_2 = Moment of inertia of the front wheels (= Iw) (kgm²)
K_1 = Spring constant, steering wheel — front wheels $\left(= \dfrac{K_w}{UTV^2} \right)$ (Nm/rad)
C_1 = Damping constant, steering wheel — front wheels (Nms/rad)
C_2 = Damping constant front wheels — body (= Cw) (Nms/rad)
RC = Radius of curvature (the general appearence of the path) (m)
AK_1 = Constant for steering on deviation
AK_3 = Constant for calculating steering wheel angle during periods of presteering
TDX = Density of controlling the steering wheel angle (s)
SLX = Time for releasing the steering wheel (s)
UTV = Steering gear ratio (—)

In this calculation Δ_w and XM are not used.

1.1.3.2 *Polynoms*

12 polynoms are fed in. There are:

1) Side force = f (slip angle) front
2) Side force = f (slip angle) rear
3) Camber angle = f (roll angle) front
4) Camber angle = f (roll angle) rear
5) Side force = f (camber angle) front
6) Side force = f (camber angle) rear
7) Roll moment = f (roll angle) front
8) Roll moment = f (roll angle) rear
9) Roll moment = f (roll angle velocity) front
10) Roll moment = f (roll angle velocity) rear
11) Self aligning torque = f (slip angle) front
12) Self aligning torque = f (slip angle) rear

Polynom 1–6, 11, 12 are per wheel.
Polynom 7–10 are per axle.

1.2 Steering model for simulation acc to ESV

1.2.1 *Mathematical Equation*

$$\ddot{\delta}_w = \frac{K_w}{I_w} \cdot (\Delta_w - \delta_w) - \frac{C_w}{I_w} \cdot \dot{\delta}_w + \frac{2}{I_w} (M + S \cdot CAS)$$

The steering wheel angle, conversed to the road wheel is:

$$\Delta_w = \Delta_{w\max} \cdot \sin^2\left(\frac{\pi}{2} \cdot \frac{t}{X_M}\right) \quad \text{for} \quad t \leq X_M$$

$$\Delta_w = \Delta_{w\max} \qquad\qquad \text{for } t > X_M$$

1.2.2 *Designations*

$\ddot{\delta}_w$ = The front wheel acceleration
$\dot{\delta}_w$ = The front wheel velocity
δ_w = The front wheel angle
Δ_w = The steering wheel angle at the front wheels
I_w = Moment of inertia, front wheels about their z-axis
K_w = Spring constant, steering wheel — front wheels (at the road wheels)
C_w = Damping constant steering wheel — front wheels
M = Self aligning torque
S = Side force
CAS = Caster offset
t = Time
X_M = Time for max steering wheel angle
$\Delta_{w\max}$ = Max steering wheel angle

1.3 Steering Model for Simulation of Controlled Driving

1.3.1 *Mathematical Equations*

$$\begin{cases} I_1 \cdot \ddot{X}_1 = K_1(X_2 \cdot i - X_1) + C_1(\dot{X}_2 \cdot i - \dot{X}_1) + F \\ I_2 \cdot \ddot{X}_2 = -K_1\, i\,(X_2 \cdot i - X_1) - C_1 \cdot i(\dot{X}_2 \cdot i - \dot{X}_1) - C_2 \cdot \dot{X}_2 + 2(M + S \cdot CAS) \end{cases}$$

X_1, X_1, X_1 are known here and we obtain:

$$\begin{cases} F = I_1 \cdot \ddot{X}_1 - K_1(X_2 \cdot i - X_1) - C_1(\dot{X}_2 \cdot i - \dot{X}_1) \\ \ddot{X}_2 = -K_1/I_2 \cdot i(X_2 \cdot i - X_1) - C_1/I_2 \cdot i(\dot{X}_2 \cdot i - \dot{X}_1) - C_2/I_2 \cdot X_2 + 2/m_2\ (M + S \cdot CAS) \end{cases}$$

When the steering wheel is released the following equations apply:

$$\begin{cases} I_1 \cdot \ddot{X}_1 = K_1(X_2 \cdot i - X_1) + C_1(\dot{X}_2 \cdot i - \dot{X}_1) \\ I_2 \cdot \ddot{X}_2 = -K_1 \cdot i(X_2 \cdot i - X_1) - C_1 \cdot i(\dot{X}_2 \cdot i - \dot{X}_1) - C_2 \cdot \dot{X}_2 + 2(M + S \cdot CAS) \end{cases}$$

1.3.2 *Designations*

I_1 = Moment of inertia of steering wheel
K_1 = Spring constant, steering wheel — front wheels
C_1 = Damping constant, steering wheel — front wheels
I_2 = Moment of inertia of road wheels about their z-axis
C_2 = Damping constant front wheels — body
i = Steering gear ratio
M = Self aligning torque per front wheel
S = Side force per front wheel

CAS = Caster offset
F = Steering wheel moment
$\ddot{X}_1$ = Steering wheel acceleration
$\dot{X}_1$ = Steering wheel velocity
X_1 = Steering wheel angle
$\ddot{X}_2$ = Road wheel acceleration
$\dot{X}_2$ = Road wheel velocity
X_2 = Road wheel angle

1.3.3 *Steering functions*

Steering angle velocity during the period of presteering

$$\dot{R}_v = K_0 \cdot UTV \cdot \frac{U}{RC}$$

$$R_v = \int_0^{t_f} \dot{R}_v(t)\,dt$$

Length of presteering period

$$X = 0.1453\;U + U^2(0.01335 - 2.212 \cdot 10^{-6} \cdot U^2)$$

The distance X is measured in the direction of the X-axis in the global system of coordinates in which the vehicle is moving.

Steering wheel angle velocity during the controlled period

$$\dot{R}_v = [K_1 \cdot UTV(Y_t - Y_v) + K_2 \cdot AV] \cdot \frac{1}{\Delta t}$$

$$R_v = \int_{t_f}^{t} \dot{R}_v\,dt + R_{vf}$$

$$K_2 = \left(\frac{4.5}{U} - 0.05\right) \cdot (1 - e^{-20 \cdot RC/U^2}) \cdot \frac{UTV}{22} \qquad \text{for} \quad /AV_{i+1}/ - /AV_i/ > 0$$

$$K_2 = 0 \qquad \text{for} \quad /AV_{i+1}/ - /AV_i/ \leq 0$$

1.3.4 *Designations*

U = Vehicle speed
RC = Radius of curvature
UTV = Steering gear ratio
t = Time
t_f = Point of time when the presteering ends
R_{vf} = Steering wheel angle at the point of time tf
Y_t = Theoretical yawing velocity (the path)
Y_v = Real yawing velocity (the car)
AV = Deviation of the center of gravity from the theoretical path
Δt = Time indicating the density of control i.e. how often the steering model acts on new information
$\left.\begin{matrix} K_0 \\ K_1 \\ K_2 \end{matrix}\right\}$ = Control constants for closed-loop system

2 Basic data for VESC

2.1 *Constant values*

1 Total vehicle mass, $m = 1529$ kg
2 Sprung mass, $m_s = 1326$ kg
3 Vehicle moment of inertia about z-axis, $I_{zz} = 2659$ kgm^2
4 Sprung mass moment of inertia about roll axis, $I_\varphi = 759$ kgm^2
5 Sprung mass product of inertia about roll and z-axis, $I_{\varphi z} = 272$ kgm^2
6 Steering wheel moment of inertia, $I_1 = 0.050$ kgm^2
7 Moment of inertia, front wheels and steering parts, $I_2 = 1.3$ kgm^2
8 Stiffness, steering wheel — front wheels, $K_1 = 25$ Nm/rad
9 Damping, steering wheel — front wheels, $C_1 = 1$ Nms/rad
10 Damping, front wheels — body, $C_2 = 100$ Nms/rad
11 Roll steer coefficient front, $\varepsilon_1 = 0.003$ rad/rad
12 Roll steer coefficient rear, $\varepsilon_2 = 0$ rad/rad
13 Track, $t = 1.48$ m
14 Wheel base, $l = 2.70$ m
15 Distance c.g. to center of front wheels, $a = 1.32$ m
16 Height of roll axis above the ground at the front axle center, $h_f = 0.054$ m
17 Height of roll axis above the ground at the rear axle center, h_r 0.331 m
18 Perpendicular distance from sprung mass c.g. to roll axis, $e = 0.38$ m
19 Steering gear ratio, $UTV = 22$
20 Rolling resistance of tires, $= 0.016$ N/N
21 Projected frontal area A = 1.95 m^2
22 Aerodynamic side force coefficient, $= 0.044$ 1/degree
23 Aerodynamic roll moment coefficient, $= 0.014$ 1/degree
24 Aerodynamic yawing moment coefficient, $= 0.007$ 1/degree
25 Aerodynamic drag coefficient, $C_x = 0.40$
26 Caster offset, $CAS = 0.016$ m

2.2 *Polynoms*

1 Side force on tires versus slip angle front wheels

$$Y_1 = 3.74 \cdot 10^4 \cdot \alpha_F - 1.93 \cdot 10^5 \cdot \alpha_F^2 + 4.10 \cdot 10^5 \cdot \alpha_F^3 \qquad \text{(N)}$$

2 Side force on tires versus slip angle rear wheels

$$Y_2 = 3.64 \cdot 10^4 \cdot \alpha_R - 1.57 \cdot 10^5 \cdot \alpha_R^2 + 3.04 \cdot 10^5 \cdot \alpha_R^3 \qquad \text{(N)}$$

3 Camber angle versus roll angle front

$$\theta_1 = 0.8 \cdot \varphi \qquad \text{(rad)}$$

4 Camber angle versus roll angle rear

$$\theta_2 = 0 \qquad \text{(rad)}$$

5 Side force on tires versus camber angle front wheels

$$Y_{1_\theta} = 1.23 \cdot 10^3 \cdot \theta_1 + 8.68 \cdot 10^3 \cdot \theta_1^2 \qquad \text{(N)}$$

6 Side force on tires versus camber angle rear wheels

$$Y_{2_\theta} = 0 \qquad \text{(N)}$$

7 Roll moment versus roll angle front

$$L_{s1} = 4.45 \cdot 10^4 \cdot \varphi \qquad \text{(Nm)}$$

8 Roll moment versus roll angle rear

$$L_{s1} = 1.01 \cdot 10^4 \cdot \varphi \qquad \text{(Nm)}$$

9 Roll moment versus roll angle velocity front

$$L_{SA1} = 1.52 \cdot 10^3 \cdot \dot{\varphi} + 6.15 \cdot 10^3 \cdot \dot{\phi}^2 - 9.99 \cdot 10^3 \cdot \dot{\phi}^3 \qquad \text{(Nm)}$$

10 Roll moment versus roll angle velocity rear

$$L_{SA2} = 1.07 \cdot 10^3 \cdot \dot{\phi} + 4.67 \cdot 10^3 \cdot \dot{\phi}^2 - 6.78 \cdot 10^3 \cdot \dot{\phi}^3 \qquad \text{(Nm)}$$

11 Wheel aligning torque versus slip angle front wheels

$$AT_1 = 5.48 \cdot 10^2 \cdot \alpha_F + 1.18 \cdot 10^3 \alpha_F^2 - 1.91 \cdot 10^4 \cdot \alpha_F^3 \qquad \text{(Nm)}$$

12 Wheel aligning torque versus slip angle rear wheels

$$AT_2 = 6.55 \cdot 10^2 \cdot \alpha_R - 2.60 \cdot 10^3 \alpha_R \qquad \text{(Nm)}$$

SECTION TWO PART 6

PROGRESS REPORT ON THE AIR BAG DEVELOPMENT FOR THE VOLVO EXPERIMENTAL SAFETY CAR (VESC)

OLOV SAXMARK, *Project Manager*
Interior Safety
AB VOLVO

In our efforts to protect vehicle occupants from injuries during the extremely severe frontal 50 mph impact in the VESC performance specification, we have chosen the following phases in the first part of our VESC air bag development program. This work is carried out as a mutual development program between Eaton S.p.A. Safety Systems Division, Bosconero, Italy and AB Volvo.

DEVELOPMENT PROGRAM

Phase 1. Design for 30 mph

Develop a basic system design for a 30 mph head-on barrier collision acceleration level.

Phase 2.

Extend the basic system to perform reliably at a 50 mph head-on barrier collision acceleration level.

Phase 3.

Check the extended basic system performance at a low speed collision acceleration level.

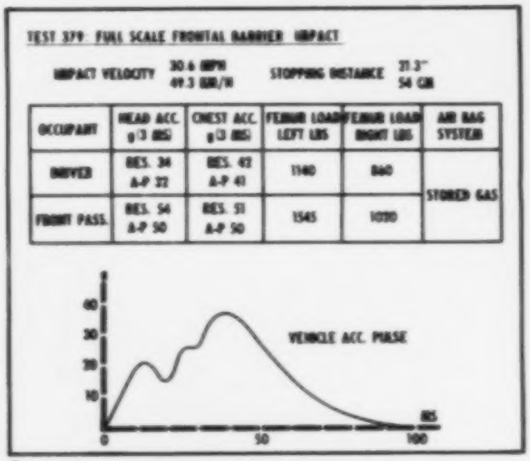
TEST DESCRIPTION AND PERFORMANCE

Phase 1. Design for 30 mph

Since barrier impact speeds up to 30 mph seem to include the usual forces developed during highway accidents, we consider the solving of problems related to collisions up to this acceleration level to be of primary importance.

Test 379 demonstrates an impact at 30.6 mph with a Volvo model 164 (Figure 1). The stopping distance is comparatively short (21.3 inches), which gives a high mean acceleration level. This level is most likely to correspond to that of the VESC at 50 mph, but obviously with a longer duration for the VESC.

In the figures RES = Resultant acceleration A-P = Anterior – Posterior acceleration measured in the head in the direction of eyeballs out, in the chest in the direction perpendicular to the spinal cord and backwards and in the pelvis in the direction of front-to-back. Dummies = 50 percentile male and 50 percentile female Sierras.



TEST 379: FULL SCALE FRONTAL BARRIER IMPACT

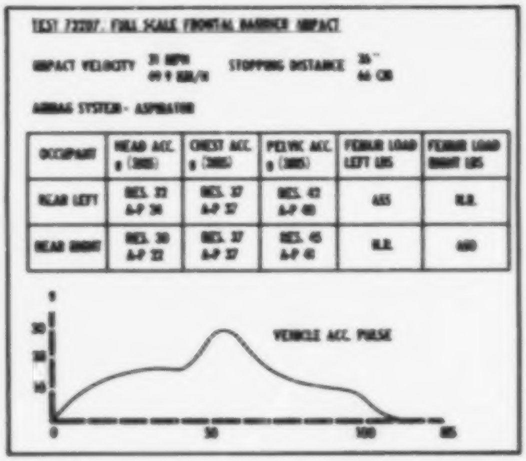
IMPACT VELOCITY 30.6 MPH 49.3 KM/H STOPPING DISTANCE 21.3" 54 CM

OCCUPANT	HEAD ACC. g (3 MS)	CHEST ACC. g (3 MS)	FEMUR LOAD LEFT LBS	FEMUR LOAD RIGHT LBS	AIR BAG SYSTEM
DRIVER	RES. 34 A-P 32	RES. 42 A-P 41	1140	860	STORED GAS
FRONT PASS.	RES. 54 A-P 50	RES. 51 A-P 50	1545	1030	

Figure 1

We are satisfied with the results obtained for the driver. However, the front seat passenger has made it with certain difficulties.

Test 73207 is a full-scale barrier impact with a Volvo model 144, which shows an entirely different vehicle pulse (Figure 2). This test was concerned only with rear seat occupants. The results are satisfactory,



TEST 73207: FULL SCALE FRONTAL BARRIER IMPACT

IMPACT VELOCITY 31 MPH 49.9 KM/H STOPPING DISTANCE 26" 66 CM

AIRBAG SYSTEM - ASPIRATOR

OCCUPANT	HEAD ACC. g (3MS)	CHEST ACC. g (3MS)	PELVIC ACC. g (3MS)	FEMUR LOAD LEFT LBS	FEMUR LOAD RIGHT LBS
REAR LEFT	RES. 33 A-P 26	RES. 37 A-P 37	RES. 47 A-P 40	455	N.R.
REAR RIGHT	RES. 30 A-P 22	RES. 37 A-P 37	RES. 45 A-P 41	N.R.	680

Figure 2

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and please observe the correspondence between the values recorded in the different dummies.

Phase 2. Design for 50 mph

Preparations for tests made during this phase were as follows. The volume of the front seat occupant bags were made twice the size of those designed for Phase 1. The amount of stored gas was increased by one third, but the pressure was kept the same, 3,500 psi. The rear seat occupant air bag restraint system remained the same design as in Phase 1, due to the promising low acceleration values registered with this system.

Test 980 was carried out with two front seat occupants and two occupants in the rear seat at a barrier impact speed of 51.7 mph (Figure 3). The stopping distance turned out to be rather long (48.8 inches), but the mean vehicle acceleration level was acceptable and high, due to the excessive speed. Front seat occupants came out well, except for an unnecessarily high-pelvis acceleration in the driver, due to a broken knee protection. The chest acceleration of the rear seat occupants were acceptable, however, in spite of a partly broken air bag system structure. The heads hit the air bag frame structure and registered high acceleration values.

Test 864 shows another frontal barrier sled impact (Figure 4). The speed was 46.2 mph and the stopping distance 48.1 inches. The acceleration figures were excellent, even if the test conditions were less severe than what is expected in a 50 mph barrier crash. The rear left dummy head experienced a very short and high vertical acceleration. This is why the RES. 70 g:s and A P 30 g:s have such different numerical values.

TEST 980 FRONTAL BARRIER IMPACT WITH SLED

IMPACT VELOCITY 51.7 MPH 83.2 KM/H STOPPING DISTANCE 48.8" 124 CM

OCCUPANT	HEAD ACC g 3MS	CHEST ACC g 3MS	PELVIS ACC g 3MS	AIR BAG SYSTEM
DRIVER	RES. 56 A-P 40	RES. 55 A-P 53	RES. 58* A-P 55	STORED GAS
FRONT PASS	RES. 55 A-P 43	RES. 60 A-P 59	RES. 44 A-P 44	STORED GAS
REAR LEFT	RES. 96 A-P 96	RES. 60 A-P 57	N.R	ASPIRATOR
REAR RIGHT	RES. 98 A-P 98	RES. 56 A-P 56	N.R	ASPIRATOR

* DRIVERS KNEE PROTECTION BROKE

NOTE REAR AIRBAG SYSTEM STRUCTURE FAILED

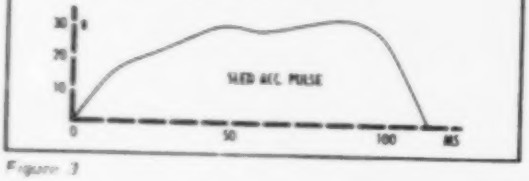

Figure 3

TEST 864. FRONTAL BARRIER IMPACT WITH SLED

IMPACT VELOCITY 46.2 MPH 74.4 KM/H STOPPING DISTANCE 48.1" 122 CM

OCCUPANT	HEAD ACC g (3MS)	CHEST ACC g (3MS)	FEMUR LOAD LEFT LBS	FEMUR LOAD RIGHT LBS	AIR BAG SYST. SAME AS IN TEST C 975
REAR LEFT	RES. 70* A-P 30	RES. 36 A-P 34	529	641	ASPIRATOR
REAR RIGHT	RES. 36 A-P 28	RES. 31 A-P 30	353	661	

* ISOLATED PEAK

NOTE EXCELLENT RESULT DUE TO RATHER "SOFT" TEST CONDITIONS

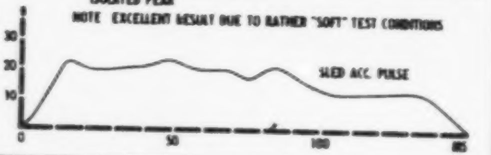

Figure 4

All other figures show good correlation between the left and the right dummy.

Phase 3. Design – same as Phase 2 – low speed

It is of importance to be able to use the same air bag system in high speed as well as in low speed impacts. We know that one can sensor one system to deploy at a low acceleration level and another system at a high acceleration level. Such a solution, however, tends to be more expensive and may introduce additional sources to produce malfunction. Accordingly, we chose to carry out Phase 3 with the same air bag system design as in Phase 2.

Low speed test C975 had an impact velocity of 19.4 mph and a 11.8 inches stopping distance (Figure 5). Registered acceleration figures were very much on the safe side and, consequently, we have succeeded in

TEST C 975: FRONTAL BARRIER IMPACT WITH SLED

IMPACT VELOCITY 19.4 MPH 31.2 KM/H STOPPING DISTANCE 11.8" 30 CM

OCCUPANT	HEAD ACC. g (3MS)	CHEST ACC. g (3MS)	PELVIC ACC. g (3MS)	AIR BAG SYSTEMS SAME AS IN TESTS 864 AND 980
DRIVER	RES. 22 A-P 22	RES. 40 A-P 40	RES. 30 A-P 27	STORED GAS
FRONT PASS.	RES. 19 A-P 17	RES. 23 A-P 22	RES. 23 A-P 22	STORED GAS
REAR LEFT	RES. N.R A-P 23	RES. 27 A-P 26	RES. N.R A-P N.R	ASPIRATOR
REAR RIGHT	RES. 35 A-P 31	RES. 31 A-P 30	RES. N.R A-P N.R	ASPIRATOR

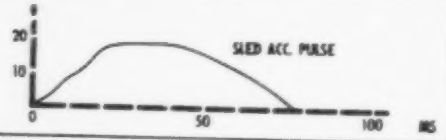

Figure 5

using the same systems for high, as well as low, speed impacts. The figures are almost equal in this test with the exception of the driver chest figures, but these are still on a reasonable level.

CONCLUSION

We have now informed you of one step in our air bag development program. The next step will repeat the described three phases but with more sophisticated designs, e.g., the front seat occupant will have an aspirator system.

We strongly believe that after the fulfillment of the next step we will have enough knowledge to equip a VESC vehicle with air bags, barrier crash it at 50 mph and get acceptable results.

The development of air bag systems is a meaningful task, but in spite of some very promising results so far, we wish in no way to underestimate the difficulties ahead of us. To mention one detail only, it may be wise to equip the VESC with complementary lap belts for additional safety in complicated traffic accidents.

THE SWEDISH TECHNICAL PRESENTATION

SECTION TWO

PART 6

A STATISTICAL TRAFFIC ACCIDENT ANALYSIS

A. ASBERG, *Head, Analysis Section*
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AB Volvo

INTRODUCTION AND ABSTRACT

This statistical analysis is based on accident data of Volvo vehicles during an approximate 12-month period completed in Sweden in 1972. The data collected in the Volvo Traffic Accident Research Project consists of comprehensive information about the accident and traffic environment, vehicle data, the occupants and the injuries sustained.

The analysis focuses on the value of various restraint items, such as safety belts, including retractor belts and head restraints. In this respect, the analysis is a follow-up of the Volvo investigation in 1967 involving more than 28,000 accidents (1).

The analysis further includes an evaluation of the material, in general, and the total accident cases, particularly in view of applicable requirements of the VESC specifications. The basic statistical calculations and estimations were made by Lars Westerlund, Chalmers University of Technology, Gothenburg.

METHOD OF DATA COLLECTION

The collection of data was made in close cooperation with Volvo dealers in Sweden and in connection with the Volvo PVG "Five-Year-Guarantee." This guarantee means, briefly, that each delivered Volvo car is insured by the company against damage, in case of an accident, during the first five years after delivery. The accident investigation and the applicable data collection were initiated when a damaged vehicle, under guarantee, was brought to the dealer for repairs. The conditions were, therefore, those which were present in the previous study (1).

The gathering of all case data, except occupant-injury data, was taken care of by special data-collectors who were situated at the dealers concerned. The data-collectors, who were employees of the Volvo company, were trained in filling out the questionnaire and interviewing the customer, in taking the required measurements of the vehicle and producing photo documentation.

The questionnaire used was prepared especially for the study and consisted of two parts (see Appendixes). The first part was a six-page information section dealing with the following groups of data:

	No. of questions
• road description and traffic situation	18
• accident type information	8
• vehicle data	54
• occupant data	13
• weather and visibility	7
• time of day identification	4

The second part, was referenced to the hospital or doctor, to whom the possibly injured occupants were brought, and included slides of the vehicle. The completed forms were then sent to the Volvo Traffic Accident Research Group. While the first part of the form was forwarded directly to the Data Computer Department for punching, the reference part was subjected to careful analysis and evaluation by a skilled accident-investigation team. The medical expert on the team requested detailed information about the injured occupant from the hospital, and the technical researcher evaluated the slides of the vehicle. The information was further discussed and analyzed by the group, entered on additional report forms and finally transferred to punched cards to join the data of the first part already in the databank.

INVESTIGATION CASE CRITERIA

The investigation criteria were to consider cases involving only:

- current models of the Volvo 140 and 164 with
- repair costs of 2,000 Swedish Crowns (U.S. $400) and above or
- any occupant injury, regardless of vehicle repair costs, or
- any other exceptional factors

The investigation was geographically limited to four main areas:

1. Stockholm and rural surroundings (Central Sweden)
2. Gothenburg and rural surroundings (West Sweden)

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(1) Bohlin, N.I., "A Statistical Analysis of 28,000 Accident Cases, With Emphasis on Occupant-Restraint Value," *SAE Transactions*, Vol. 76, No. 670925.

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3. Malmo and rural surroundings (South Sweden)
4. Sundsvall and rural surroundings (North Sweden)

ANALYZED MATERIAL

The material collected and analyzed was put in relation to the total number of "Five-Year Guarantee" cases in Sweden for the period concerned, and is summarized below. Reference is made in terms of estimated repair costs, which are meant to reflect the severity of accidents in general.

REPAIR COSTS (S U.S.) NUMBER OF CASES

	$ 400	$ 400 $1,400	$1,400 $1,800	Over $1,800	Totals
Total in Sweden	27 684	8 556	994	491	37 725
Districts Concerned	7 251	2 808	115	113	10 287
Material Analyzed	270	920	115	98	1 403 + 102 [illegible]

There is a significant difference in the distribution of repair costs between the material collected and analyzed and the total numbers of accident cases in Sweden (guarantee cases). The difference is that the material collected has more high-cost repair cases than the total. This overrepresentation is primarily related to the most expensive cases.

Percentage of material costs in Sweden
(total of FYG-cases)

Repair costs U.S. $ 400-$1,400 10.7%
Repair costs U.S. $1,400-$1,800 11.6%
Repair costs U.S. over $1,800 20%

The number of low-cost cases (non-severe accidents), which is not the subject of this analysis, forms the main part (73%) of the total number of accidents in Sweden.

Of the accidents "available" during the collecting period in the districts concerned, the material of the study shows:

Repair costs U.S. $ 400-$1,400, 33% (approximate)
Repair costs U.S. $1,400-$1,800, 100%
Repair costs U.S. over $1,800, 82% (approximate)

BASIC COLLECTED DATA

The material analyzed comprises 1,505 accidents which comply with the criteria mentioned. A breakdown of the 2,440 occupants involved, is as follows:

- 1,505 drivers (61.5%). 1,473 vehicles had left-side steering
- 503 front seat passengers (20.7%)
- 432 rear seat passengers (17.8%)

The analysis of occupant-restraint value in the report deals with front-seat occupants in vehicles with left-side steering.

VEHICLE DEFORMATION INDEX (VDI)

The deformation of the case vehicles was codified firstly, according to the non-linear scale in the "Collision Deformation Classification SAE J224a" and, secondly, according to the linear scale suggested by Volvo in the *U. S. Pilot Study on Traffic Accident Investigations*. In the linear scale, the front half, as well as the rear half, of the vehicle and the total width from one side to the other side are divided into 10 equal parts. To simplify the summary in this report, the results refer only to the non-linear scale.

CLASSIFICATION OF INJURIES

The injuries were codified according to AIS (Abbreviated Injury Scale) (See Appendix I).

ACCIDENT TYPES

The accident types in the material were divided into three main groups, depending on how the accident occurred:

- collision with another vehicle or object
- running off the road
- rollover on the road surface

Quite often there was a combination of two or even all three of these. The full classification of the types is shown as:

• total number of accidents	1,505
• number of classified cases into accidents	1,474
• number of unknown types	31

Collisions (C)
(total 1239)

C	C + O	C + O + R	C + R	C + R + R
1195	27	8	8	1

Running off the road (O) (total 229)

O	O + C	O + C + R	O + R	O + R + C
19	160	12	23	15

Rollover (R)
(total 6)

R	R + O	R + O + C	R + C	R + C + O
1	0	1	4	0

C = Collision
O = Running off the road
R = Rollover

The *impact directions* in the total number of accidents, when related to a clock diagram were found to be as follows (see also Figure 19):

frontal	(12 o'clock)	34%
frontal	(11 o'clock)	16%
frontal	(01 o'clock)	14%
left side	(10 o'clock)	3%
left side	(09 o'clock)	4%
left side	(08 o'clock)	1%
rear	(07 o'clock)	2%
rear	(06 o'clock)	12%
rear	(05 o'clock)	2%
right side	(04 o'clock)	1%
right side	(03 o'clock)	5%
right side	(02 o'clock)	3%
roof		1%
unknown		2%

The *collision* accident was the most dominating type. Among them, the "pure collision," i.e. some kind of impact without running off the road or rollover, occurs most frequently.

• impact (beginning of accident)	1,239 cases
• impact following running off the road	187 cases
• impact following rollover	5 cases

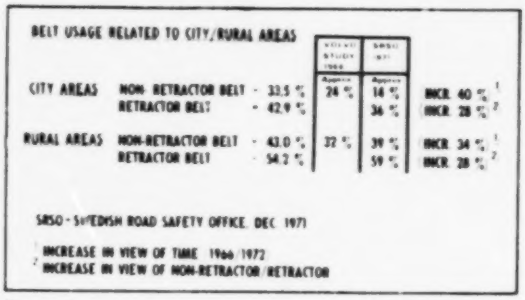
The other two main types of accident – running off the road and rollover – represented, together, the smaller part (15.7%) of the total.

USE OF SAFETY BELTS

All case vehicles were equipped with Volvo's 3-point slip-joint belts in the front seat. In the rear seats, safety belts of the 3-point type for the outer seats and lap-type belts for the center seat appeared in approximately 60% of the cases.

Front seat belt use: The overall use of safety belts by driver and front passenger was 39.2%. The mean use of the non-retractor belt in city areas (shorter trips and usually a 30-mph speed limit) proved to be 33.5%, an improvement of 40% of the result (approximately 24%) in the 1967 Volvo study (1). The corresponding figures for rural areas (longer trips and higher speed limits) were 43%, an improvement of approximately 32% in relation to the 1967 study (Figures 1 and 2).

Rear seat belt use: Only 22 (5.1%) of the 432 rear-seat passengers used a seat belt. Of the 432, however, only 282 were travelling in belt-equipped vehicles. The relative use increases, therefore, to 7.8%. There was no evident difference between the three rear seats.



BELT USAGE RELATED TO CITY/RURAL AREAS

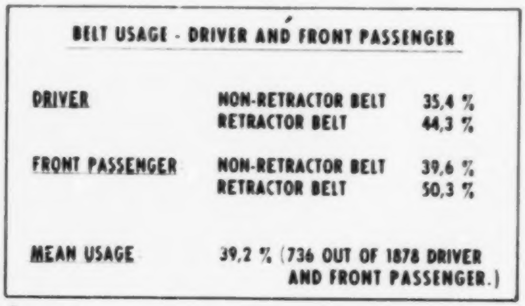
			VOLVO STUDY 1966 Approx.	SRSO 1971 Approx.	
CITY AREAS	NON-RETRACTOR BELT	- 33.5 %	24 %	14 %	INCR. 40 % [1]
	RETRACTOR BELT	- 42.9 %		36 %	INCR. 28 % [2]
RURAL AREAS	NON-RETRACTOR BELT	- 43.0 %	32 %	39 %	INCR. 34 % [1]
	RETRACTOR BELT	- 54.2 %		59 %	INCR. 28 % [2]

SRSO - SWEDISH ROAD SAFETY OFFICE, DEC. 1971

[1] INCREASE IN VIEW OF TIME 1966/1972

[2] INCREASE IN VIEW OF NON-RETRACTOR/RETRACTOR

Figure 1



BELT USAGE - DRIVER AND FRONT PASSENGER

DRIVER	NON-RETRACTOR BELT	35,4 %
	RETRACTOR BELT	44,3 %
FRONT PASSENGER	NON-RETRACTOR BELT	39,6 %
	RETRACTOR BELT	50,3 %
MEAN USAGE	39,2 % (736 OUT OF 1878 DRIVER AND FRONT PASSENGER.)	

Figure 2

BELT USE RELATED TO OCCUPANT AGE

As may be seen from Figure 3, the percentage of use increased with increasing occupant age, from a mean of 30 – 35% at 20 years of age to 50 – 55% at approximately 50 – 55 years of age.

USE OF RETRACTOR VS NON-RETRACTOR BELTS

Of the belt cases, 28.5% involved a retractor connected to the upper torso strap. Of the drivers, 44.5% took advantage of their retractor-equipped belts, i.e., an improvement of 25% over the use of non-retractor belts. The front-seat passengers were

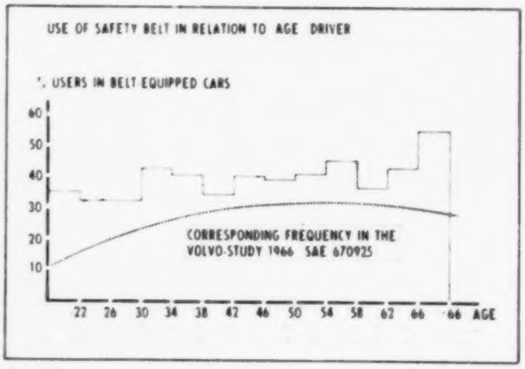


Figure 3

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(1) Ibid.

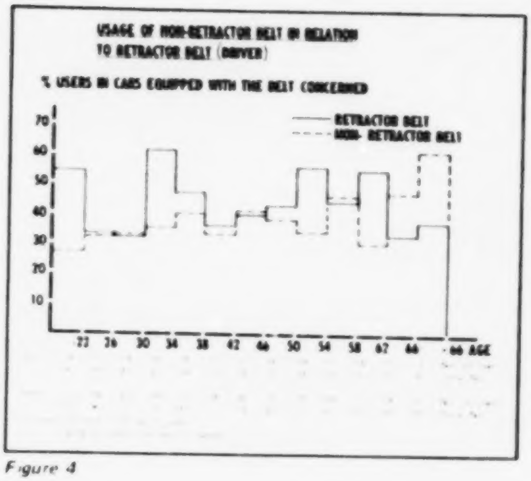

Figure 4

still more cooperative (50.3%), an improvement of 30%. The frequency of use of the retractor belt showed no noticeable relationship to age distribution (Figure 4). In fact, the retractor belt showed a lower use frequency, as compared with the non-retractor belt for the age of 62 years. It might be explained simply by the comparatively small number of observations, or it could be the effect of pressure of the upper torso strap on the chest, which older people could find irritating.

HEAD RESTRAINTS

The effect of head restraints, in terms of the number of neck injuries sustained, was evaluated in rear-end collisions, which were specified to impact directions 04 08 on the clock diagram.

In the 171 rear-end collisions, the case vehicles were equipped with head restraints for the front seats in 73.6% (126). The severity of the rear impact is depicted in the Vehicle Deformation Index (Figure 5). The main part is referred to VDI = 1 group (116). Vehicle deformations with VDI = 2 and 3 are reported in 30 cases. The two groups vehicles *with* head restraints and vehicles *without* head restraints show roughly similar VDI-representation. To reassure complete information of any neck injury or hazard, including those appearing some days after the accident and therefore not reported, all occupants involved were contacted and interviewed.

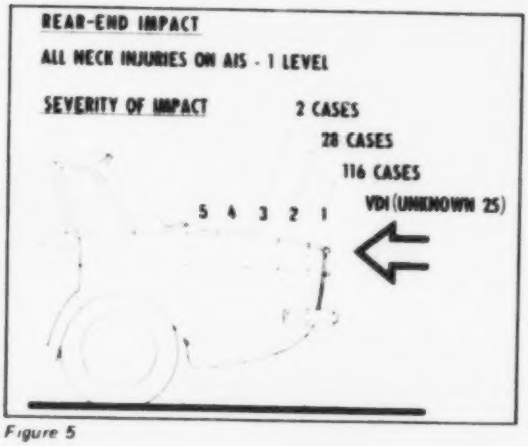

Figure 5

FREQUENCY OF NECK INJURIES RELATED TO USE OF HEAD RESTRAINT (DRIVER SEAT)

REAR END IMPACT DIRECTION 04 - 08

HEAD RESTRAINT 20 INJURED OUT OF 126 = **15,9 %**

NO HEAD RESTRAINT 16 INJURED OUT OF 45 = **35,3 %**

INJURY-REDUCING EFFECT OF HEAD RESTRAINT = 55%

Figure 6

Due to the small totals, when the material is divided into VDI and AIS degrees, especially on the >1-degree level, the material does not permit a meaningful evaluation in terms of AIS related to VDI, but only to the frequency of neck injuries.

Head restraint cases showed a significantly lower frequency of neck injuries 15.9% (20 out of 126) than no-head restraint cases 35.3% (16 out of 45). Significance level was 0.05 (Figure 6).

The neck injuries sustained were, however, not very severe in either group. There was only AIS-1 degree injury, i.e. "whiplash" complaints (pain or strain) with no anatomical or radiological evidence. The injury symptoms appeared immediately after the accident in 50% of the cases, after one day in 30%; and after two days in 20% of the cases.

The fact that there were no severe neck injuries in the no-head restraint cases could possibly be credited to the yielding features of the back rest of the front seats. The reclining device of the seat is of a friction type, which is preset to yield for a certain load applied to the top of the back rest. The preset friction is high enough to withstand normal loading but low enough to yield in an excessive acceleration situation.

THE EFFECT OF THE SAFETY BELT – GENERAL

The injury-reducing effect of safety belts tells to what degree the number of injured occupants using belts is less than the number of injured occupants not using belts. The effect is given in the percent of the number of unbelted occupants. No reference to severity degree of injuries sustained is made.

The evaluation of the safety-belt effect refers to front and rear seat occupants, in left-hand steering vehicles only. The effect of safety belts in the front seats is referred both to conventional (non-retractor) 3-point belts and 3-point safety belts with retractor for upper torso strap.

Two types of *statistical methods* were used to analyze the effect of safety belts; partly tests and partly calculation of confidence intervals. Tests have been used to determine if the belt effect is significant in various accident situations. Confidence intervals for p-values — i.e. the probability to sustain injuries in various conditions — have further been calculated. The 95% intervals are marked on graphs which reflect the uncertainty of the estimations. Certain statistical signs, which will not be dealt with in detail in this report, indicate, however, that the true uncertainty is somewhat less than what is shown by the intervals, i.e. the lengths of them should probably be shorter. Even though the intervals are overlapping each other, it is believed that the relations shown between the p-values (the columns) are right in most cases, even when significance is not achieved. From the discussion above, it is further noted that the uncertainty depicted by the confidence intervals is transferred to a corresponding degree of uncertainty in the estimations of the injury-reducing effect. As mentioned before, the injury-reducing effect calculated does not consider injury severity but only the number/frequency.

If the injury-reducing effect in different situations (VDI) are compared, the effect seems to be larger in a minor/moderate impact than in a more severe accident. Since the severity-degree of the injuries (AIS) is found to increase with increasing VDI, the smaller effect, in terms of quantity, is believed to be well compensated for in terms of quality.

By and large, the severity-degree of injuries sustained by belted occupants is lower than that of unbelted occupants, especially within the AIS-index groups 4–7, i.e. serious to fatal injuries, where only one single case (AIS 5) is related to belted occupant (retractor belt). In this case, the resulting true injury-reducing effect of safety belts should be higher than indicated by the figures in the various diagrams.

THE EFFECT OF BELT IN FRONT SEATS

Driver (Summary figures):

Belted
total number = 528
injured = 92 (17.4%).

Unbelted
total number = 887
injured = 226 (25.5%)

The *mean* injury-reducing effect of the safety belt on the driver was thus found to be 32%, or somewhat lower than in the previous study (1). One factor which might explain the difference is that the injury classification in this study is more rigorous and "picks up" many trifling injuries (strain, muscle ache, abrasions, etc.), which could have been characterized as "no injury" in the previous study. Even if valid for both categories, this factor is believed to have greater influence on the belt cases, which were more stringently investigated. Another explanation is that the current study refers to a basic material which represents by and large more severe accidents (more than U. S. $400, estimated repair costs).

Front seat passenger (Summary figures):

Belted
total number = 208
injured = 48 (23.1%)

Unbelted
total number = 281
injured = 102 (36.3%)

Dropout cases in this group of material, due to factors like limitation to left-side steering, type of belt unknown, use of belt unknown, were:

driver = 87 cases
front passenger = 14 cases

The *mean* injury-reducing effect for the front-seat passenger was 36%, somewhat lower than in the 1967 study, and it is referred to in the comments above for this tendency.

The mean figures above tell further that the hazard to the front-seat passenger, compared with the driver, is approximately 42% greater. This difference in hazard also is noticeable when a safety belt is used, but is then decreased to approximately 33%.

THE EFFECT OF BELT IN REAR SEATS

Belted
total number = 22
injured = 3 (13.6%)

Unbelted
total number = 410
injured = 103 (24.8%)

Very few (22) of the 282 rear-seat occupants who had had the possibility to be restrained took the advantage. The totals in this study group is too small to permit any confirmed conclusions on the belt effect. The three injuries reported are, however, all minor within AIS 1.

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(1) Ibid

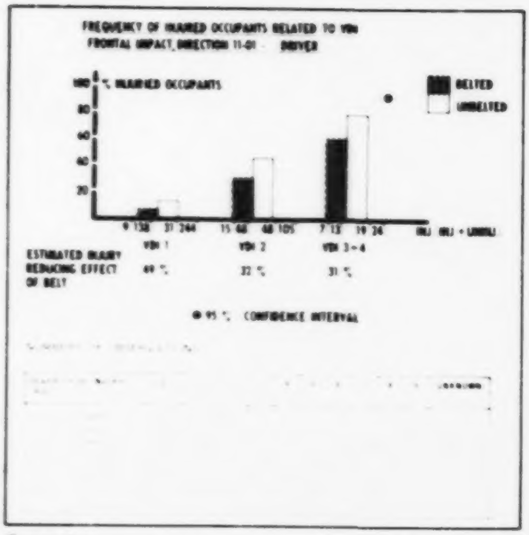


Figure 7

The injuries sustained by the unbelted rear-seat occupants comprises, on the other hand, most categories within the AIS-scale from AIS 1 up to two cases of AIS 6–7 (fatal injuries).

THE EFFECT OF SAFETY BELT RELATED TO DIRECTION OF IMPACT

To evaluate the belt effect in view of direction of impact, four sectors of impact direction were chosen:

- frontal impact direction 11–01 (ref. to clock-figures)
- left side impact direction 08–10
- right side impact direction 02–06
- rear end impact direction 05–07

The material in these four groups represent possible "clean" impact cases. That means that the accidents involved mainly a pure impact. Accidents concerning very complex accident situations, as well as cases where the distinction of impact direction (e.g. 11 front or 11 side) was not sure, were deleted. The number of observations are given in the summary tables and frequency of injuries sustained are depicted for each group below.

FRONTAL IMPACT (11–01) – (Figure 7)

Driver. The total effect of the safety belt weighed over all VDI is significant on level 0.01. The lengths of the 95% confidence intervals marked in the graphs, show that the estimation of the injury-reducing effect is much more confirmed in VDI-1 group than in VDI 3 + 4 cases. The belt effect, which varies from 49% to 31% (approximately), decreases with increasing VDI.

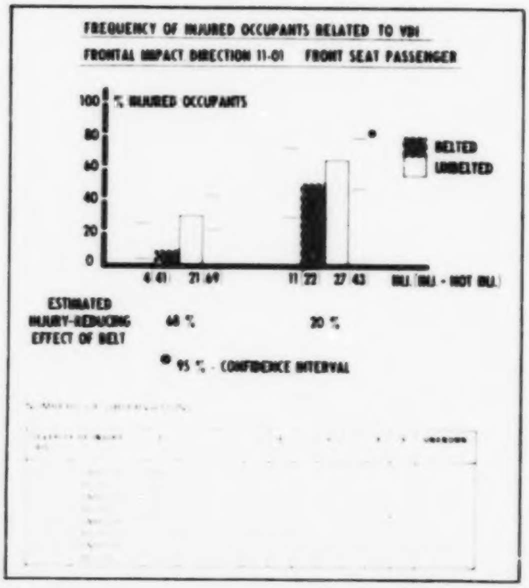


Figure 8

Front seat passenger (Figure 8). Again, the total effect of safety belt weighed over all VDI is significant on level 0.01.

The injury-reducing effect for the front-seat passenger is higher (68%) than for driver (49%) in low VDI impacts, but drops 20% in the higher VDI-numbers.

The confidence in the effect estimation is somewhat greater in VDI 1.

DRIVER RELATED TO FRONT SEAT PASSENGER (BELTED/UNBELTED)–(Figures 9-10)

For *belted* driver, compared with belted front-seat passenger, no significant difference of injury frequency could be shown.

When both driver and front-seat passenger are *unbelted,* however, a significant difference of injury frequency (level 0.01) is found to the passenger's disadvantage.

RETRACTOR VERSUS NON-RETRACTOR BELT–(Figure 11)

The figures of injury frequency for the retractor belt related to the identical non-retractor belt reveal that there is no or very little difference (not significant). This result is valid for both the driver and front-seat passenger positions.

REAR END IMPACT (05–07)–(Figure 12)

There seems to be a certain positive belt effect (24% – 35%) in rear-end impacts. The effect, which

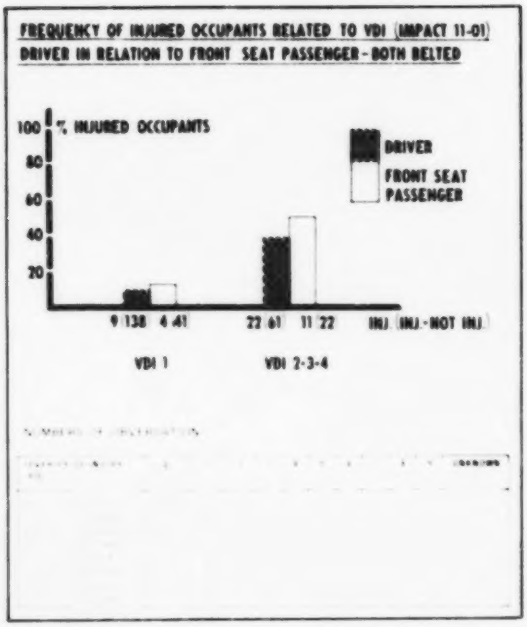


Figure 9

by and large is less than in frontal impacts is, however, not significant for either the driver or the passenger. No significant difference between driver or passenger, belted or unbelted, in terms of injury frequency could be shown.

SIDE IMPACTS (LEFT AND RIGHT) (08–10, 02–04)–(Figures 13-14)

A positive effect of the driver's belt in left-side impacts (8-10) is found to be significant (level 0,05).

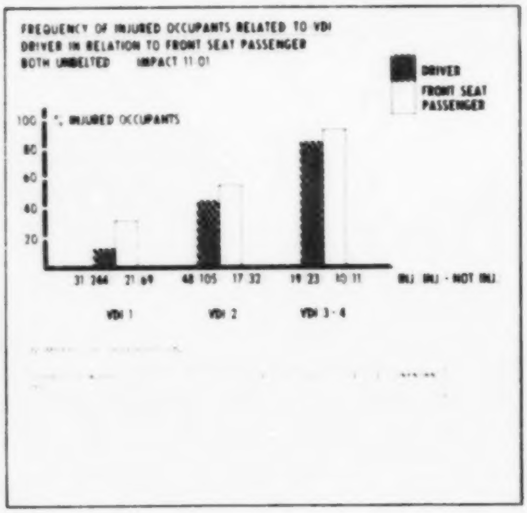


Figure 10

Figure 11

For all other side-impact parameters (driver in 02-04 impacts, passenger in 08-10 as well as 02-04 impacts) the belt effect proved to be of little significance. A reasonable explanation for the distinct difference between the left and right-side impact effect on the driver cannot be derived from the analysis carried out. The question will be subjected to further studies.

INJURIES SUSTAINED IN DEGREE OF SEVERITY

Injuries sustained by the front-seat occupants were related to AIS-scale as follows: (totals)

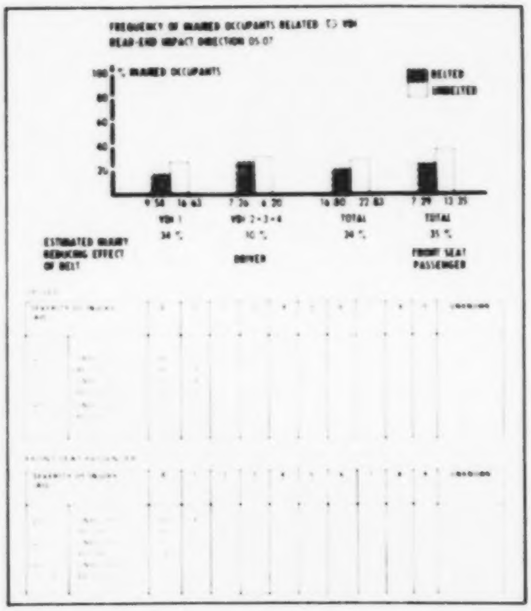


Figure 12

AIS	0	1	2	3	4	5	6	7	99X
Unbelted	840	228	15	14	1	1	2	2	65
Belted									
– Retractor	208	26	3	4	–	1	–	–	6
– Non-retractor	388	72	5	7	–	–	–	–	16

As depicted in the following graphs (Figure 15), the front-seat passenger sustained injuries of various AIS-index to higher frequency than the driver. That happened in all situations – unbelted or belted – but in the minor injury class when a safety belt without a retractor had been used there was no difference. AIS-difference was particularly dominant in the AIS 2 – 3 class, where the passenger was injured approximately two times (unbelted) and up to three times (belted with non-retractor belt) as often as the driver. Credit for the better outcome for the driver is given to the steering wheel and steering column with its "restraining" features.

All but one case of the very serious-to-fatal injuries were related to non-belted occupants. The *single critical injury*, when the belt had been properly used, was a front-seat passenger with a retractor belt in a complex accident situation.

In the accident concerned, the case vehicle impacted another vehicle, ran off the road and into a ditch, where it collided and finally was thrown against and impacted a pole with the roof. It is believed that the front-seat passenger's upper torso

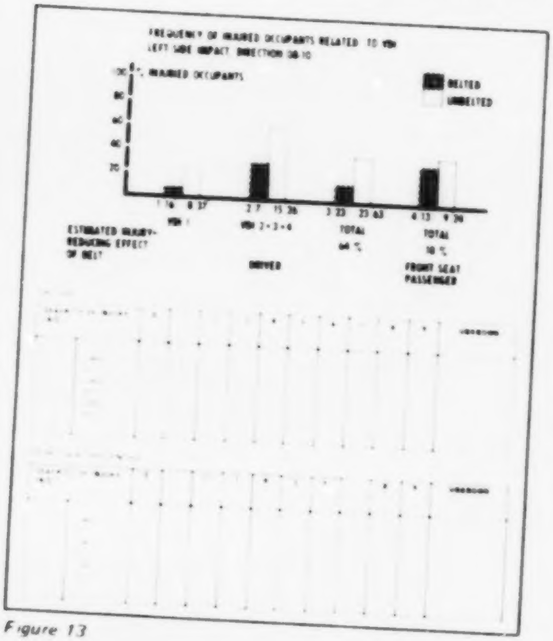


Figure 13

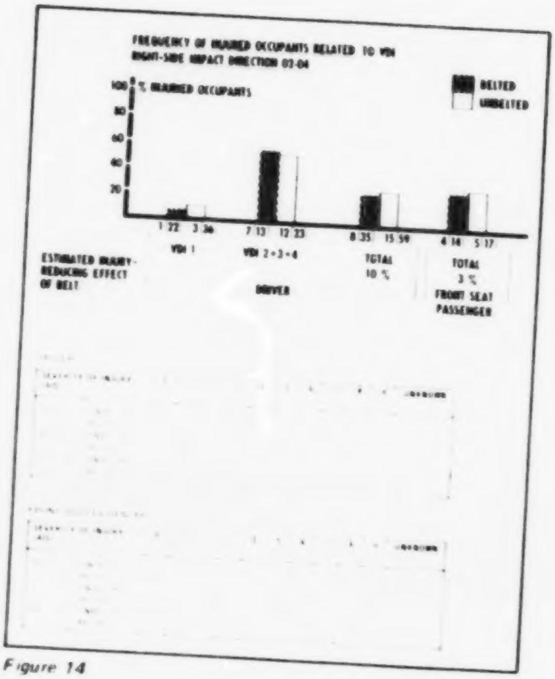


Figure 14

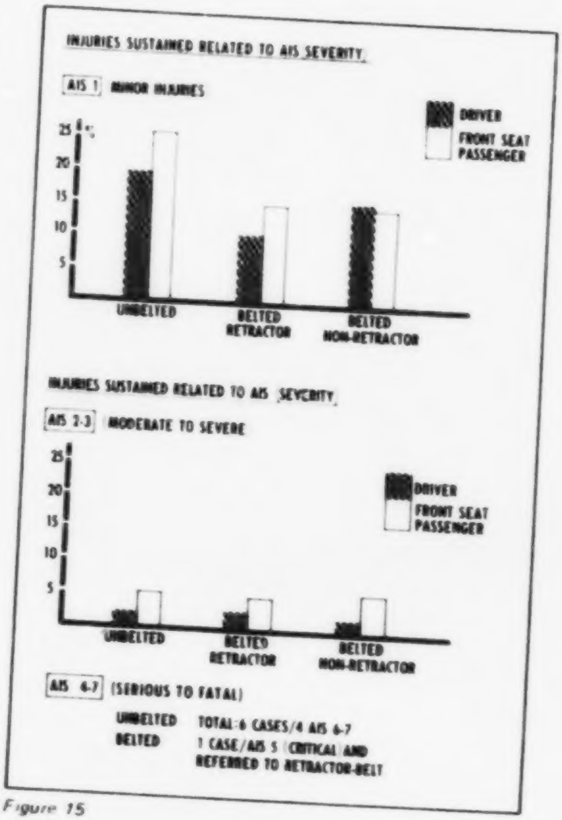


Figure 15

during the first part of the accident sequence might have slid out of the chest strap of the belt, resulting in decreased retraint efficiency during the later accident phase.

The passenger wearing a retractor belt sustained the following injuries:

- nasal fracture
- facial laceration
- concussion with loss of memory for five days (AIS 5)

The driver, also wearing retractor belt, sustained AIS-2 degree injury with scapula fracture (right) and pain in left arm.

The distribution of main injuries on the body is summarized in the diagrams in Figures 16-17.

The frequency of head/skull injuries, which is significantly dominant both for unbelted driver (18%) and unbelted front-seat passenger (43%), is cut down to 8% resp 14% when belted. No significant difference in frequency of chest injuries was found between belted and unbelted occupants, but a significant difference in severity was found. The unbelted cases were responsible for all chest injuries with AIS-severity higher than 2.

The number of leg and arm injuries were, overall, comparatively small and of minor severity. The number of AIS-3 class very small.

EJECTION

Ten occupants were ejected in eight accidents, of which all involved complex accident sequences, roll-over, rotation, etc. Five of the ejected occupants, none of whom used the safety belt, were in the front seat and five in the rear seat. The frequency of ejection counted from the total occupancy was approximately 0.4%.

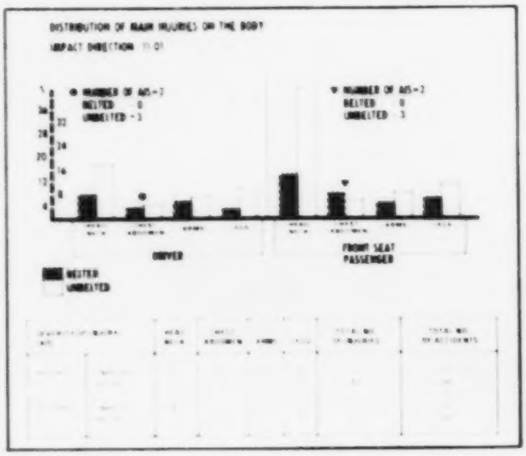
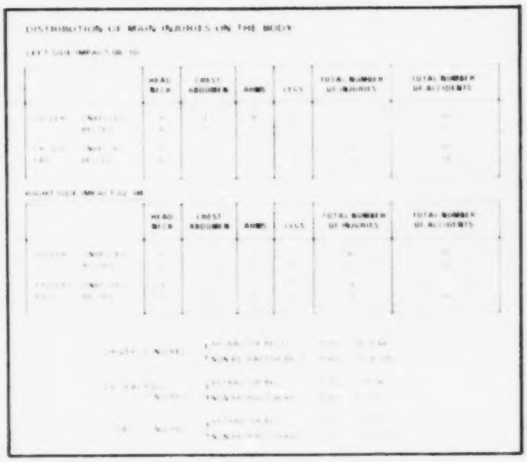


Figure 16



DISTRIBUTION OF MAIN INJURIES ON THE BODY
HEAD NECK
CHEST ABDOMEN
ARMS
LEGS
TOTAL NUMBER OF INJURIES
TOTAL NUMBER OF ACCIDENTS

Figure 17

Injuries sustained by the 10 ejected persons:

- two persons were fatally injured (AIS 6) with complex head/skull injuries.
- one person sustained critical head injuries (AIS 5).
- two persons sustained AIS 3 injuries related to thoracic spine, head area.
- four persons sustained minor injuries (AIS 1) only (slight concussion, head laceration and abrasions).
- one person was lucky to sustain no injury.

Ejection from the vehicle proved to be much more hazardous than non-ejection (unbelted). The ejected persons were killed in 20% (and injured in 90%), while the non-ejected, unbelted occupants were fatally injured only in approximately 0.3% (some injury in 26.8%). It is further noted that a substantial part of the total killed (33% 2 out of 6) is found among the ejected.

THE MATERIAL ANALYZED VERSUS THE VESC CRASHWORTHINESS SPECIFICATION AND OTHER SAFETY ITEMS

The Cases In General

The number of cases of the frontal impact group (11-01) related to Vehicle Deformation Index (VDI) were:

	Accident cases	Front seat occupant cases
VDI 1 =	382	492
VDI 2 =	153	201
VDI 3 =	25	33
VDI 4 =	9	14
VDI 5 =	1	1
VDI 6 =	2	3

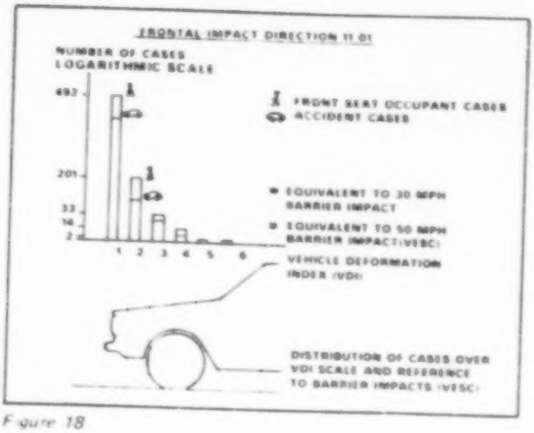


Figure 18

According to experience with extensive barrier collision tests with the vehicle models concerned, the VDI 3-accidents are judged to be approximately equivalent to the 30-mph frontal barrier impact test (Figure 18). The evaluation revealed that most frontal collision cases (98% — 560 of 572) corresponded approximately to barrier collision speeds up to 30-mph, while only about 2% of them corresponded to higher collision speeds. In other words, as the material analyzed is found to be over represented with high cost (more serious) accidents in view of the total accident pattern in all Sweden — as pictured in the Volvo PVG-statistics (five-years guarantee) — the relevancy and ambition to keep the 50-mph-barrier test specification in VESC, which in this respect is almost identical to the ESV, could be questioned. On the other hand, the belted occupants in these "50-mph-barrier"-accidents were fairly successful.

The Fatal Cases

Five accident cases resulted in six fatally-injured occupants. They were carefully analyzed and weighed in view of the questions (see Appendix 2) and described in the Volvo-report: "Fatal Accidents During a Twelve-Month Period (1972) in Volvo 140 and 164 Cars (2)," which includes all six fatalities. In this report, only a summary with estimations are given.

The fatal accidents comprised two frontal impacts (VDI 3 and VDI 6), two rollovers and one of another type. In four of the five cases, both an "improved interior" — Question 1, and "100% belt use" — Question 2 — were estimated to have a large fatal-injury-reducing effect. For the fifth accident, however, the 100% belt use was judged only to have a large positive effect.

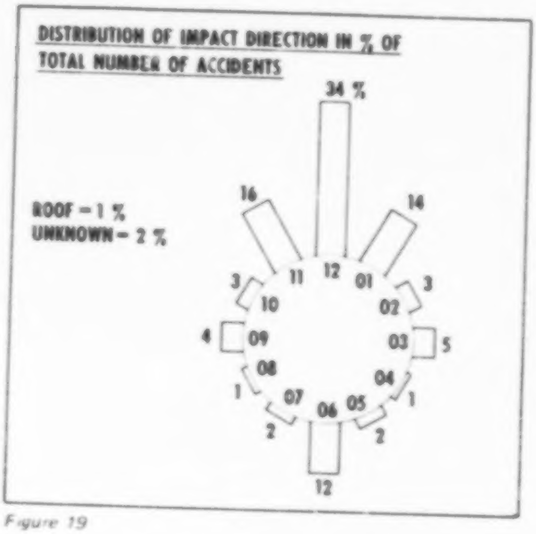

DISTRIBUTION OF IMPACT DIRECTION IN % OF TOTAL NUMBER OF ACCIDENTS
34 %
ROOF = 1 %
UNKNOWN = 2 %
16
14
3
11
12
01
3
10
02
4
09
03
5
08
04
1
07
06
05
1
2
2
12

Figure 19

The VESC specification applicable (Questions 3–6) would not have meant a large, positive, effect to any of the killed occupants. The estimations clearly indicated that the injury-reducing effect of a case vehicle compliance with the VESC-requirements would have been small or fairly small in the accidents concerned.

SUMMARY AND CONCLUSIONS

From the analysis made of the study material, which mainly represented the fourth quarter of a general composition of accident severity in terms of increased vehicle repair costs, the following is summarized and concluded:

- the *mean use of non-retractor belt* in front seats in city and rural areas was 33,5% and 43%, respectively.
- the corresponding *mean use of retractor belt* was 42,9% and 54,9%, respectively, or an increase of approximately 28% (mean) when related to non-retractor belts.
- the *use-of-belt figures for non-retractor belt* reveal an increase as high as 40% (city areas) compared with a similar study reported in 1967 (1). The credit for this improvement is given to improved belt design and continued "education" of customers.
- Use of rear seat belts was only 7,8% (mean).
- The mean injury-reducing effect of belt for driver

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(2) Samuelsson, L. I., "Fatal Accidents During a Twelve-Month Period (1972) Involving Volvo Models 140 and 164 Vehicles." AB Volvo.

(1) Op. Cit.

and front-seat passenger was 32% and 36%, respectively.

- None of the fatally-injured occupants (6) was restrained by belts.
- The resulting true effect of belt use was concluded to be still higher, since the belt evidently decreased the severity of the injury sustained.
- No or very little difference (not significant) was found between retractor and non-retractor belt use, either in terms of frequency or severity.
- The hazard to the front-seat passenger, in case of an accident, was 42% higher than to the driver when unbelted; 33% higher when belted.
- Ejection was found to be most hazardous to occupants, with 33% of the total killed, compared with a frequency of 0.4% of total occupancy.
- Head restraints in the front seats proved to be effective in reducing the frequency of neck injuries by approximately 55%.
- The severity of neck injuries was all minor (AIS 1), even in cases without head restraints. It was concluded that this probably was due to the designed yielding features of the front-seat backrests.
- From the severe accident composition of the material, it was concluded that most accidents (98% – approximately 560 of 572) corresponded to barrier collision speeds up to 30-mph, while only about 2% corresponded to higher collision speeds.
- From a detailed analysis of the fatal cases it was concluded that compliance with the VESC crashworthiness specification would have had some positive effect.

APPENDIX I

Injury Classification According To The AIS (Abbreviated Injury Scale)

Severity Code 0 – NO INJURY

Severity Code 1 – MINOR

General

- aches all over
- minor lacerations, contusions, and abrasions (first aid – simple closure)
- all 1° or small 2° or small 3° burns

Head and neck

- cerebral injury with headache, dizziness, no loss of consciousness
- "whiplash" complaint with no anatomical or radiological evidence
- abrasions and contusions of occular apparatus (lids, conjunctiva, cornea, uveal injuries); vitreous or retinal hemorrhage
- fracture and/or dislocation of teeth

Chest

- muscle ache or chest wall stiffness

Abdominal and pelvic contents

- muscle ache; seat belt abrasion; etc.

Extremities

- minor sprains and fractures and/or dislocation of digits.

Severity Code 2 – MODERATE

General

- extensive contusions; abrasions; large lacerations; avulsions (less than 3" wide)
- 10 - 20% body surface 2° or 3° burns

Head and neck

- cerebral injury with or without skull fracture, less than 15 minutes unconsciousness
- undisplaced skull or facial bone fractures or compound fracture of nose
- lacerations of the eye and appendages; retinal detachment
- disfiguring lacerations
- "whiplash" – severe complaints with anatomical or radiological evidence

Chest

- simple rib or sternal fractures
- major contusions of chest wall without hemothorax or pneumothorax or respiratory embarrassment

Abdominal and pelvic contents

- major contusion of abdominal wall

Extremities and/or pelvic girdle

- compound fractures of digits
- undisplaced long bone or pelvic fractures
- major sprains of major joints

Severity Code 3 – SEVERE (Not Life-Threatening)

General

- extensive contusions; abrasions; large lacerations involving more than two extremities, or large avulsions (greater than 3" wide)
- 20 – 30% body surface 2° or 3° burns

Head and neck

- cerebral injury, with or without skull fracture, with unconsciousness more than 15 minutes, without severe neurological signs, brief posttraumatic amnesia (less than three hours)

- displaced closed-skull fractures without unconsciousness or other signs of intracranial injury
- loss of eye, or avulsion of optic nerve
- displaced facial bone fractures or those with antral or orbital involvement
- cervical spine fractures without cord damage

Chest

- multiple rib fractures without respiratory embarrassment
- hemothorax or pneumothorax
- rupture of diaphragm
- lung contusion

Abdominal and pelvic contents

- contusion of abdominal organs
- extraperitoneal bladder rupture
- retroperitoneal hemorrhage
- avulsion of urethra
- thoracic or lumbar spine fractures without neurological involvement

Extremities and/or pelvic girdle

- displaced simple long-bone fractures, and/or multiple hand and foot fractures
- single open long-bone fractures
- pelvic fracture with displacement
- dislocation of major joints
- multiple amputations of digits
- lacerations of the major nerves or vessels of extremities

Severity Code 4 – SERIOUS (Life-Threatening)

General

- severe lacerations and/or avulsions with dangerous hemorrhage
- 30–50% surface 2° or 3° burns

Head and neck

- cerebral injury, with or without skull fracture, with unconsciousness of more than 15 minutes, with definite abnormal neurological signs; post-traumatic amnesia 3–12 hours
- compound skull fracture

Chest

- open-chest wounds, flail chest, pneumomediastinum, myocardial contusion without circulatory embarrassment, pericardial injuries

Abdominal and pelvic contents

- minor laceration of intra-abdominal contents (to include ruptured spleen, kidney and injuries to tail of pancreas)
- intraperitoneal bladder rupture
- avulsion of the genitals
- thoracic and/or lumbar spine fractures with paraplegia

Extremities

- multiple-closed long-bone fractures
- amputation of limbs

Severity Code 5 – CRITICAL (Survival Uncertain)

General

- over 50% body surface 2° or 3° burns

Head and neck

- cerebral injury, with or without skull fracture, with unconsciousness of more than 24 hours; post-traumatic amnesia of more than 12 hours; intracranial hemorrhage; signs of increased intracranical pressure (decreasing state of consciousness, brady-cardia under 60, progressive rise in blood pressure or progressive pupil inequality)

Chest

- chest injuries with major respiratory embarrassment (laceration of trachea, hemomediastinum etc.)
- aortic laceration
- myocardial rupture or contusion with circulatory embarrassment

Abdominal and pelvic contents

- rupture, avulsion or severe laceration of intra-abdominal vessels or organs, except kidney, spleen or ureter

Extremities

- multiple open-limb fractures

Severity Code 6 – FATAL (Within 24 hours)

- fatal lesions of single region of body, plus injuries of other body regions of Severity Code 3 or less; fatal from burns regardless of degree.

Severity Code 7 – FATAL (Within 24 hours)

- fatal lesions of single region of body, plus injuries of other body regions of Severity Code 4 or 5.

Severity Code 8 – FATAL

- two fatal lesions in two regions of body

Severity Code 9 – FATAL

- three or more fatal injuries
- incineration by fire

Severity Code 10 – FATAL

- death, but details unknown

Severity Code 99 X – SEVERITY UNKNOWN

- injured, but severity not known

Severity Code 98 Z – PRESENCE UNKNOWN

- presence of injury not known

APPENDIX 2

METHOD OF ANALYSIS OF FATAL ACCIDENTS IN VIEW OF VESC-SPECIFICATION

The analysis method used here involved using the documentation of each accident to answer a number of pre-arranged questions and then to evaluate these answers against specific answer options.

QUESTIONS

1. What fatal injury-reducing effect would interior improvements have had, concerning an energy-absorbing steering wheel and steering column, fixed energy-absorbing front and rear seats and a passive-protection system in the form of padding, air bags or similar items?

   (The passive-protection systems mentioned here include systems which do not securely restrain the occupant throughout the entire accident.)
2. What fatal injury-reducing effect would a 100% use of safety belts and children's safety seats have had? (Appraisal has not taken into consideration any loads carried on the rear seat etc.)
3. What fatal injury-reducing effect would a VESC-body have had with regard to frontal collisions?
4. What fatal injury-reducing effect would a VESC-body have had with regard to lateral collisions?
5. What fatal injury-reducing effect would a VESC-body have had with regard to collisions from the rear?
6. What fatal injury-reducing effect would a VESC-body have had with regard to rollover accidents? (In the appraisal of items 3 – 6, consideration was taken as to how the actual accident conformed with the conditions in the VESC requirements.
7. How many accidents could have been avoided if the vehicles had been equipped with anti-skid brakes? (In this appraisal, consideration was given to whether the driver of the car braked with locked wheels, braking distance length and the possibilities, with unlocked wheels, for driver to steer to avoid the collision obstacle.)

All answers have been graded according to the following scale

1. Large positive effect
2. Fairly large positive effect
3. Fairly small positive effect
4. Slight positive effect
5. Not applicable.

APPENDIX 3

FATAL ACCIDENTS DURING A TWELVE-MONTH PERIOD (1972), INVOLVING VOLVO MODELS 140 AND 164 VEHICLES

LARS SAMUELSSON,
Traffic Accident Research
AB Volvo

INTRODUCTION

This report is only concerned with collisions involving Volvo model 140 and model 164 vehicles in Sweden during 1972. The conclusions made are relevant within the framework of these limits.

PURPOSE

The purpose of the investigation was to indicate the situations, and the ways occupants are fatally injured in Volvo 142, 144, 145 and 164 model vehicles, and to what degree, currently known, safety-improvement items could have led to a reduction in the number of fatal injuries. The items which have been evaluated are: improved interior with energy-absorbing units, safety belts and the VESC-body, with regard to impact energy absorption in frontal, lateral, rear and roof deformation, and also anti-skid brakes.

DATA

The material analyzed consists of the total number of fatal accidents in Volvo 140 and 164 cars and is based on official fatal accident reports. A fatal accident is considered an accident where, as a consequence of injuries sustained, an occupant dies within 30 days of the accident.

DOCUMENTATION

Police reports, photographs of the vehicles and autopsy report death certificates are available for all the accidents included in this report. Some of the accidents have been examined by Volvo's expert group.

ROAD TRAFFIC REVIEW IN SWEDEN IN 1972

All figures, with the exception of the number of fatal accidents in Volvo cars and Volvo's market share, are expolations. The total number of cars was 2,400,000, of which approximately 240,000 were Volvo 140's and 164's. The accidents totaled 16,500 and involved 15,500 cars; 23,000 individuals were injured and 650 killed.

Thirty-eight persons were killed in 31 Volvo 140's in 29 accidents. There were no fatal accidents in Volvo 164's.

The fatal-accident involvement was as follows:

19 frontal collisions with 23 killed
6 lateral collisions with nine killed
4 rollover accidents with four killed
2 other accidents with two killed

ANALYSIS METHOD

The analysis method consisted of using the documentation of each accident to answer a number of prearranged questions and then to evaluate these answers against specific answer options.

QUESTIONS

1. What fatal injury-reducing effect would interior improvements, involving an energy-absorbing steering wheel and steering column, fixed energy-absorbing front and rear seats and a passive-protection system in the form of padding, air bags or similar items, have had?

   (The passive-protection systems mentioned here include systems which do not securely hold the occupant throughout the entire accident).
2. What fatal injury-reducing effect would a 100% use of safety belts and children's safety seats have had?

   (Appraisal has not taken into consideration any loads carried on the rear seat, etc.).
3. What fatal injury-reducing effect would a VESC body have had with regard to frontal collisions?
4. What fatal injury-reducing effect would a VESC body have had with regard to lateral collisions?
5. What fatal injury-reducing effect would a VESC body have had with regard to collisions from the rear?
6. What fatal injury-reducing effect would a VESC body have had with regard to rollover accidents? (In the appraisal of Items 3 – 6, consideration was given as to how the actual accident conformed with the conditions in the VESC requirements, which are listed below.)
7. How many accidents could have been avoided if the vehicles had been equipped with anti-skid brakes?

   (In this appraisal, consideration was given to whether the driver of the car braked with locked wheels, braking distance, and the possibility of steering past the collision obstacle with unlocked wheels.)

All answers were graded according to the following scale:

1. Large positive effect
2. Fairly large positive effect
3. Fairly small positive effect
4. Slight positive effect
5. Not applicable
0. Unknown

VOLVO EXPERIMENTAL SAFETY CAR (VESC)

Definition of requirements

Max 3 inches compartment intrusion at 50-mph barrier front collision

Max 3 inches compartment intrusion at 50-mph front collision pole

Max 3 inches compartment intrusion at 50-mph front collision barrier 15°

Max 3 inches compartment intrusion at 50-mph front collision pole 15°

Max 3 inches compartment intrusion at 30-mph front collision barrier 45°

Max 3 inches compartment intrusion moving barrier rear collision 50-mph

Max 3 inches compartment intrusion moving pole rear collision 50-mph

Max 3 inches compartment intrusion moving barrier rear collision 15° 50-mph

Max 3 inches compartment intrusion moving pole rear collision 15° 50-mph

Max 3 inches compartment intrusion vehicle to vehicle rear collision 75-mph

Max 4 inches compartment intrusion lateral collision 30 – 40-mph

Max 3 inches compartment intrusion lateral collision rigid pole 15-mph

Max 3 inches roof intrusion

The VESC requirements, against which these accidents were rated, correspond to the U. S. ESV requirements for similar situations, with the exception of the lateral collision where a 4-inch intrusion is accepted in the VESC project.

BASIS OF EVALUATION

Evaluation was made from two different viewpoints, one with the focus on the accident vehicle and the other with the focus on the killed occupant.

The purpose of focusing on the vehicle was to avoid any random, or irrelevant, factors involved in the accident. For this reason, the accident situation elements – vehicle deformation and the reduction in size of the passenger compartment – are the factors on which appraisal is primarily based.

The accident vehicles can be roughly divided into three groups: a) passenger compartment undamaged; b) passenger compartment reduced in size but VESC body would have kept it undamaged; and c) where vehicle deformation and passenger-compartment reduction greatly exceeded the plausible limits of safety design. For accident vehicles within group a) an improved interior, or, optionally, a 3-point belt has, in principle, been judged as having large or fairly large fatal injury reducing effect, while the VESC body has been judged as having a low, or fairly low, effect. For accident vehicles in group b) the VESC-body has been judged as having a large, or fairly large, fatal-accident reduction effect. The total of groups a) and b) forms the total fatal-accident reduction effect for the VESC-body, with improved interior, and, also, for the VESC-body with 100% use of safety belts.

The purpose of focusing on the killed occupant was to indicate how many lives could have been saved. For this reason, the reduction of the size of the passenger compartment reported here is related to the seat which the deceased occupied in the vehicle. Otherwise, the factors analyzed conform with those analyzed for the vehicle. Independent evaluation has been made for interior improvements and for 100% use of safety belts.

RESULTS – CONCLUSIONS

(Figure 1)

On the basis of the Volvo 140-series cars, it would have been possible to achieve a fatal-injury reduction of between 40% and 55% through interior improvements, alone, or, optionally, through 100% use of safety belts.

An improvement in interior design, along the lines mentioned previously, would have resulted in a fatal injury reduction in 12 of 31 vehicles. Eleven of these vehicles were involved in a head-on collision where the passenger compartment was undamaged or only slightly damaged. One case involved was a side impact towards a pole. (The lateral collisions which resulted in fatal accidents are not covered by the VESC requirements. In two cases, trucks travelling at speeds exceeding 50 km/h hit cars from the side, another case involved a bus travelling at about 50 km/h, another involved a train travelling at 110 km/h and three involved vehicles which skidded off the road at very high speed and side-impacted a pole or tree.)

The appraised results of 100% use of safety belts coincide with results attained from an improved interior, in the case of head-on collisions. The increase in fatal-injury reduction shown for 100% use of safety belts is the result of rollover accidents (4), a type of accident in which only safety belts have been judged to be effective. In these cases the passenger compartment was undamaged or only slightly damaged. In one case, an eight year old boy lying on the rear shelf was ejected through the rear window. In a second case, the driver, travelling alone, was thrown to the opposite side of the car where his head crashed through the side window and was crushed between the roof and the road. In another case, a three year old girl was ejected through the door aperture and in one accident a 72 year old woman was tumbled round in the vehicle and broke her neck. Totally, the use of the safety belt alone could have saved lives in 17 of 31 vehicles.

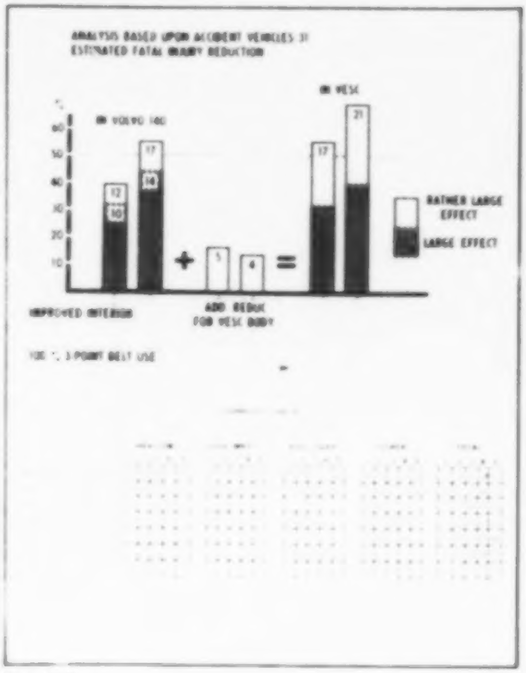


Figure 1

If it were supposed that the VESC requirements applying to body design (see above) had been conformed with in the accident vehicles, some additional reduction would have been gained (between 13% and 16%). These cases consist of an accident where two Volvo 142's collided head-on, at exactly "12 o'clock," at high speed (over 80 km/h) and also two accidents where the frontal collision was slightly off-side (11 o'clock). In all of these vehicles, considerable passenger compartment reduction was noted, but for the two latter, it was not of vital importance in the implications of the accident. In addition to this, one case was involved where it was presumed that the VESC-body would have had a fairly large positive effect through the anti-intrusion (side-impact members) and the improved interior.

(Figure 2)

When analysis is based on the killed occupant, the effect of the interior improvements or, optionally, use of the safety belts, increases for the Volvo 140 vehicles.

The fatal injury reduction through interior improvements or optional 100% use of safety belts together with the VESC-body is compatible in both the vehicle and occupant analysis.

In conclusion it can be said that 100% use of safety belts in the Volvo 140 is judged to give the same fatal injury reduction as the Volvo Experimental Safety Car without safety belts but with passive, non-restraining, protective systems.

The VESC-body alone, with the current interior and without safety belts, would have given a fatal injury reduction in one accident vehicle.

If the vehicles involved had been equipped with an anti-skid brake system, three accidents could have been avoided.

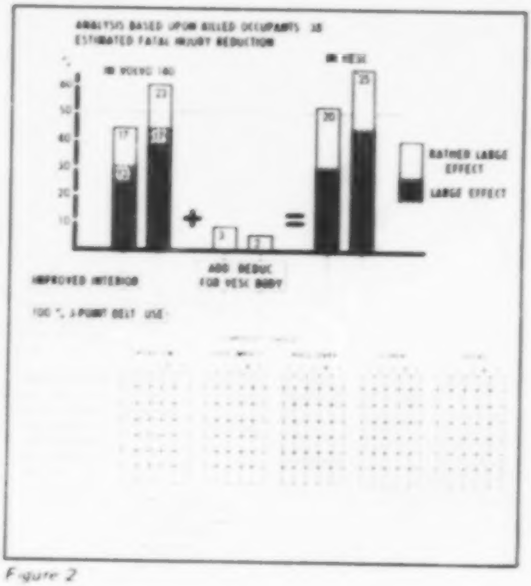
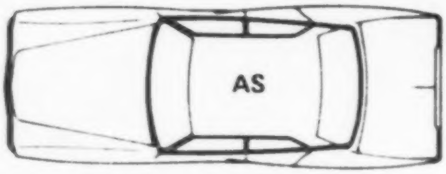


Figure 2

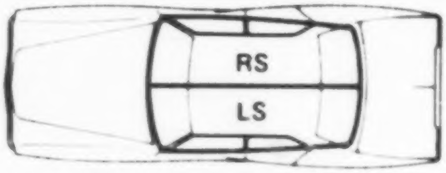
APPENDIX 4
ACCOUNT FOR THE EVALUATION OF EACH ACCIDENT

VEHICLE INTERIOR DEFORMATION INDEX

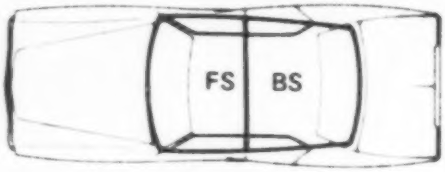
col 1 – 2



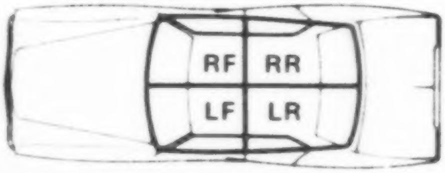
AS ALL SEATS



RS RIGHT SEATS
LS LEFT SEATS



FS FRONT SEATS
BS REAR SEATS



LF LEFT FRONT
LR LEFT REAR
RF RIGHT FRONT
RR RIGHT REAR

D = DRIVER
FP = FRONT PASSENGER
RP = REAR PASSENGER

An index has been drawn up for each individual accident (VDI covering collision deformation classification SAE 3224 a) and also for passenger compartment reduction as per the following

VIDI is partly based on the same principles as the VDI and is meant to be used in combination with the VDI

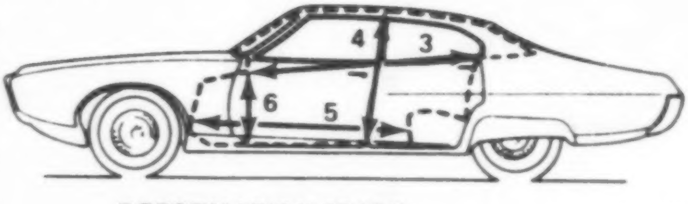
VDI gives a detailed specification of the location and extent of external deformation. VIDI is meant to complete this information regarding the reduction of the passenger compartment.

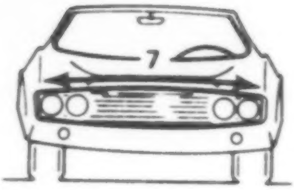
The amount of reduction is given in a 10-degrees linear scale with "0" meaning no reduction, "5" meaning reduction to the half and "X" meaning total reduction

The location of the reduction is related to the seat/seats concerned.

Col 3 7	are used to indicate the reduction, where
Col 3	means the deformation between the top of the rear seat back rest and the instrument panel (not overhanging padding) or intruding objects
Col 4	means the deformation between floor and roof
Col 5	means the deformation between the foremost lower part of the rear seat and the fire wall
Col 6	means the deformation between the lower part of the instrument panel and the floor
Col 7	means the inner width deformation.

col 3 – 7





VIDI

DEFORMATION SCALE

0 1 2 3 4 5 6 7 8 9 X

INDEX COLUMN NO- 1 2 3 4 5 6 7

ACCIDENT NO. 1

Volvo 142/1971. Type of accident: car underran truck.

Evaluation

1. Personal injuries due to contact with steering wheel.
   D 1
2. Examination of passenger compartment after accident indicates that correctly used safety belt would have had effect.
   D 1
3. Of no importance in this type of underrun accident.
   D 4
4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. 65 m long skid marks from locked wheels.
   D 1

Remarks: Driver, sole occupant, killed.
Cause of death: chest injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: unknown, but probably not used.
Collision speed: below 50 km/h, estimated on extent of deformation VDI 12FRXA6, VIDI. 0.

Accident No. 1

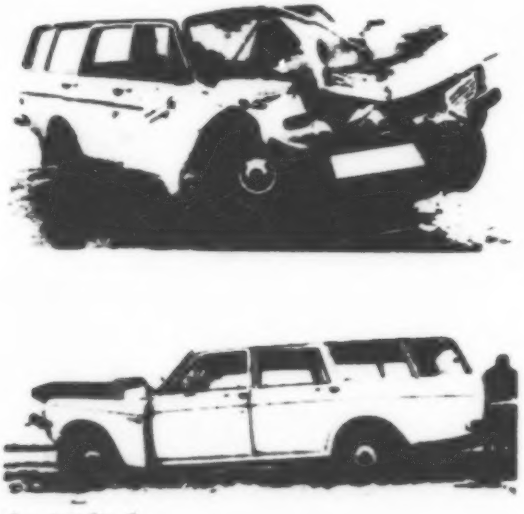
ACCIDENT NO. 2

Volvo 145/1969. Type of accident: head-on collision with road grader parked on highway.

Evaluation:

1. Rear-seat passenger, without seat belt, thrown forward onto driver causing extensive deformation of seat backrest and crushing driver to death against steering wheel and safety belt.
   D 1
2. Without unbelted rear seat passenger, safety belt would have had effect.
   D 1
3. Accident speed below 50 km/h. Passenger compartment largely intact.
   D 4
4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. 102.5 m long skid marks from locked wheels.
   D 1

Remarks: Four occupants in accident vehicle. Driver killed; belted. Front-seat passenger seriously injured (AIS 5) crushed against dashboard; not belted. Rear-seat passenger slightly injured; not belted.
Cause of death: chest injuries.
Joint appraisal of occupants and accident vehicle.
Safety belt: used by driver.
Collision speed: below 50 km/h estimated on extent of deformation. VDI 12 FZEW 3, VIDI FS10100.



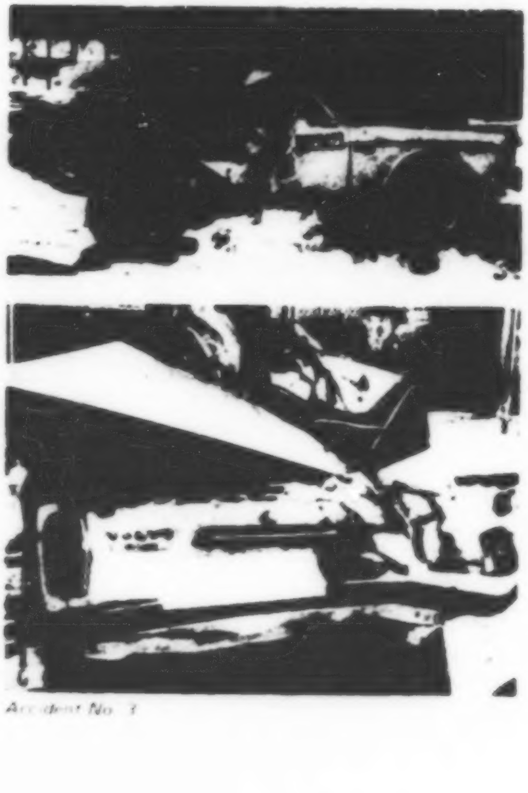
Accident No. 2

ACCIDENT NO. 3

Volvo 144/1972. Type of accident: skidded with rightside foremost against approaching truck of normal-control type.

Evaluation:

1. Severity of accident (see Item 4) was of such extent that influence of interior design can be disregarded.
   D 4
2. Driver was thrown from left to right side; can be presumed that had seat belt been used, would have had effect.
   D 2
3. Not applicable.
   D 5
4. Speed of skidding car was between 70 – 90 km/h, speed of approaching truck was 70 km/h, implying that conditions of VESC requirement were exceeded.
   D 3
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5



Accident No. 3

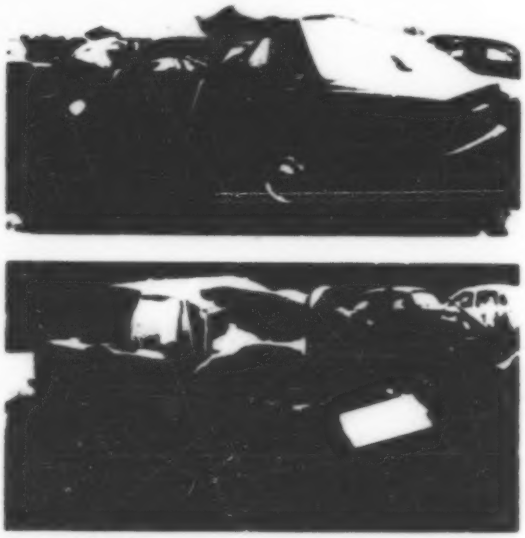
Remarks: Driver, sole occupant, killed.
Cause of death: head injury.
Joint appraisal of occupant and accident vehicle.
Safety belt: unknown, but probably not used.
Collision speed: Volvo 144, 70 – 90 km/h, truck 70 km/h; based on witnesses and time-speed recorder.
VDI 03RZAW5, VIDI RS00005.

ACCIDENT NO. 4

Volvo 142/1970. Type of accident: car underran truck.

Evaluation:

1. Severity of accident (see Item 3 below) was of such extent that influence of interior design can be disregarded.
   D 4
2. Use of safety belt would have had very limited effect, due to intrusion by truck platform and extensive deformation of the roof.
   D 3
3. Of no importance in this type of accident. Underrunning a stationary truck; speed 90 – 100 km/h
   D 4
4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5



Accident No. 4

Remarks: Driver, sole occupant, killed.
Cause of death: head injury.
Joint appraisal of occupant and accident vehicle.
Safety belt: not used.
Collision speed: approximately 100 km/h; based on witness reports.
VDI 2FDXA7. VIDI FS31390.

ACCIDENT NO. 5

Volvo 142/1970. Type of accident: off center frontal collision with approaching vehicle.

Evaluation:

1. Personal injuries caused by front seat passenger being crushed between dashboard and seat back-rest, forced forward by dog weighing approximately 40 kg. Force of weight from rear is judged to have had considerable effect.
   FP 1, Car 1
2. Passenger compartment area for front seat passengers was intact, therefore, correctly worn seat belts would have had effect.
   Passenger compartment in driver's seat was somewhat reduced, but is thought that safety belt also would have had effect here.
   FPI, Car 1
3. Passenger compartment on passenger side entirely intact, on driver's side, same reduction took place.
   FP 4, Car 2
4. Not applicable.
   FP 5, Car 5
5. Not applicable.
   FP 5, Car 5
6. Not applicable.
   FP 5, Car 5
7. Not applicable.
   FP 5, Car 5

Accident No. 5

Remarks: Front seat passenger died as result of injuries sustained in accident; driver probably died as result of acute cerebral hemorrhage, which could also explain the accident.
Cause of death: front seat passenger: chest injuries.
Driver: intracranial bleeding.
Joint appraisal of occupant and accident vehicle.
Safety belt: not used.
Collision speed: Volvo, 142 80 km/h, approaching car, 90 km/h; based on witness reports.
VDI 12FYEW6. VIDI LF20230.

ACCIDENT NO. 6

Volvo 142/1972. Type of accident: run into on rightside by normal-control truck.

Accident No. 6

Evaluation:

1. Severity of accident was of such extent that influence of interior design can largely be disregarded.
   D 4, FP 3, RP 3, Car 3
2. Driver was probably using safety belt. Quotation taken from autopsy report: "Diagonally across the front of the chest can be seen, from top left to bottom right, a number of hemorrhages approximately the size of a pinhead . . . Lower down, from the stomach across the hips, is a section where the outer skin is abraded and the cutis is dried out and reddish brown in color."
   Not possible to say whether passengers were wearing safety belts, but use of safety belt in this

type of accident is of lesser importance.
D 3, FP 3, RP 3, Car 3
3. Not applicable.
D 5, FP 5, RP 5, Car 5
4. VESC requirements could have had some positive effect.
D4, FP 3, RP 3, Car 3
5. Not applicable.
D 5, FP 5, RP 5, Car 5
6. Not applicable.
D 5, FP 5, RP 5, Car 5
7. Not applicable.
D 5, FP 5, RP 5, Car 5

Remarks: Three occupants of car killed.

Cause of death: driver: chest and abdominal injuries, front seat passenger: chest and abdominal injuries rear seat passenger: chest injuries.

Joint appraisal of occupant and accident vehicle.

Safety belt: used by driver.

Collision speed: intruding truck, above 50 km/h.

VDI 03PRAW4, VIDI RS00004.

ACCIDENT NO. 7

Volvo 145/1969. Type of accident: run into on left side by approaching vehicle; thereafter, thrown against guide rail and rolled over.

Evaluation:

1. Driver and front seat passenger used safety belts and escaped serious injury.
RP 4, Car 3
2. Three-year old girl sitting in rear seat was ejected from car when it rolled over and her head was crushed between car and road. Children's safety seat would probably have prevented this.
RP 1, Car 1
3. Not applicable.
RP 5, Car 5
4. Extensive side deformation (10 - 11 o'clock).
RP 4, Car 2
5. Not applicable.
RP 5, Car 5
6. No reduction of passenger compartment with regard to roof deformation.
RP 4, Car 4
7. Not applicable.
RP 4, Car 5

Remarks: Three-year-old girl sitting in rear seat was thrown out through right-hand rear door and her head was crushed between car and road when car rolled over. Children's safety seat not installed.

Cause of death: rear seat passenger: head injuries.

Joint appraisal of occupant and accident vehicle.

Safety belt: (not indicated).

Collision speed: Volvo 145, 90 km/h; other car, about 70 km/h.

VDI 11LYES3 1 2RDES1 00TDS01. VIDI LS 21333.

Accident No. 7

ACCIDENT No. 8

Volvo 144/1970. Type of accident: skidded against guide rail and was hit by approaching truck.

Evaluation:

1. Steering wheel and dashboard intact; otherwise, vehicle was total wreck rearwards of front seats.
D 4, Car 4
2. Correctly used seat belt would have had some effect for driver. With regard to this accident, effect would have been lower, due to extent of deformation.
D 2, Car 3
3. Head-on impact did not cause any reduction of passenger compartment; therefore, VESC-requirements would not have had any effect.
D 4, Car 4
4. Initial impact of truck was from rear offside (5 o'clock); therefore, side impact (anti-intrusion) would have had some effect.
D 3, Car 3

5. VESC requirement 80 km/h from rear is not wholly applicable, but would probably have had effect.
   D 3, Cat 3
6. Not applicable.
   D 5, Cat 5
7. Not applicable.
   D 5, Cat 5

Remarks: Driver, sole occupant, killed, ejected from vehicle.

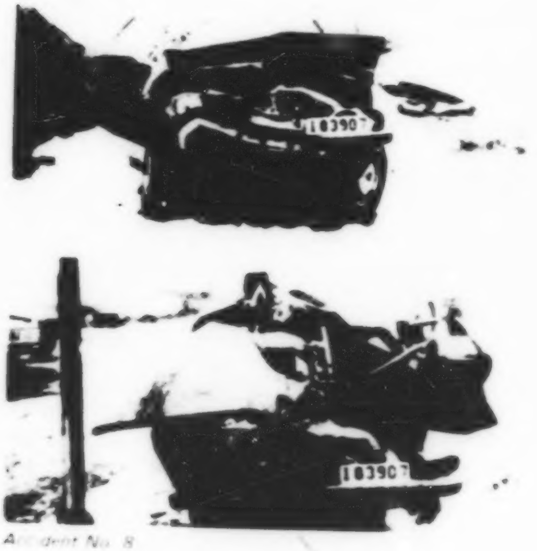
Cause of death: driver – head injuries.

Joint appraisal of occupant and accident vehicle:

Safety belt: not used.

Collision speed: Volvo 144, approx. 80 km/h; truck approx. 70 km/h.

VDI 12FDMN1 05RZAW8, VIDI BS5220S.



Accident No. 8

ACCIDENT NO. 9

Volvo 142 1969. Type of accident: offside frontal collision with cliff (1 o'clock); thereafter, continued at 120° angle to left for 10–15 meters. Speed probably over 50 km/h.

Evaluation:

1. Personal injuries due to contact with steering wheel.
   D 1, Cat 1
2. Correctly used safety belt would have had effect.
   D 1, Cat 1
3. VESC requirement 50 km/h against barrier would have had some effect.
   D 4, Cat 3
4. No applicable.
   D 5, Cat 5
5. Not applicable.
   D 5, Cat 5
6. Not applicable.
   D 5, Cat 5
7. Not applicable.
   D 5, Cat 5

Remarks: Driver, sole occupant, killed.

Cause of death: chest injuries.

Joint appraisal of occupant and accident vehicle:

Safety belt: probably not used.

Collision speed: unknown, but estimated on extent of deformation as over 50 km/h.

VDI 01FZEW5, VIDI RS20330.

Accident No. 9

ACCIDENT NO. 10

Volvo 142 1970. Type of accident: head-on collision (12 o'clock) with approaching vehicle (142 case 11).

Note: Possibility of acute heart attack having taken place immediately previous to accident can, according to autopsy report, not be ruled out. Accident was still appraised along lines of model.

Evaluation:

1. Personal injuries due to contact with dashboard and steering wheel.
   D 2
2. Correctly worn safety belt would have had effect.
   D 2

3. VESC requirements applicable; collision speed about 80 km/h.
   D 1
4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5

Remarks: Driver, sole occupant, probably dead before accident, due to heart attack.

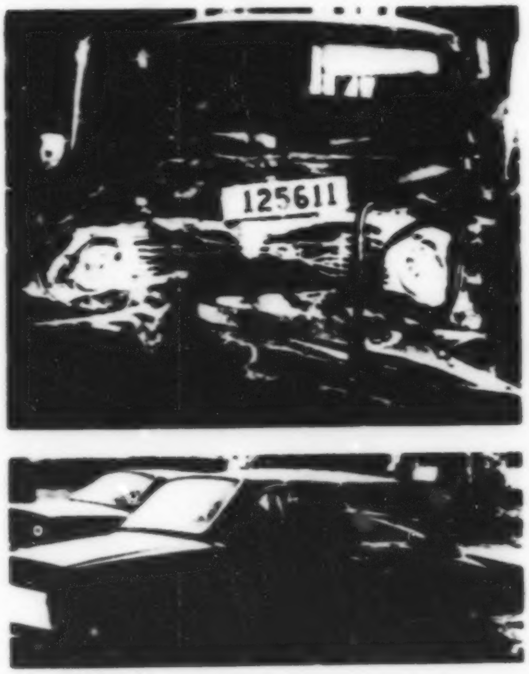
Cause of death: head and chest injuries.

Joint appraisal of occupant and accident vehicle.

Safety belt: probably not used.

Collision speed: over 80 km/h; estimated on extent of deformation.

VDI 12FDEW5, VIDI FS30350.



Accident No. 10

ACCIDENT NO. 11

Volvo 142/1971. Type of accident: head-on collision (12 o'clock) with approaching vehicle (142 case 10).

Evaluation:

1. Personal injuries due to contact with steering wheel. Seven-year-old girl thrown forward and sustained crushed skull.
   D 2, RP 2, Car 2
2. Correctly used safety belts would have had effect.
   D 2, RP 2, Car 2
3. VESC requirements applicable; collision speed about 80 km/h.
   D 1, RP 4, Car 1
4. Not applicable.
   D 5, RP 5, Car 5
5. Not applicable.
   D 5, RP 5, Car 5
6. Not applicable.
   D 5, RP 5, Car 5
7. Not applicable.
   D 5, RP 5, Car 5

Remarks: Driver and rear seat passenger killed.

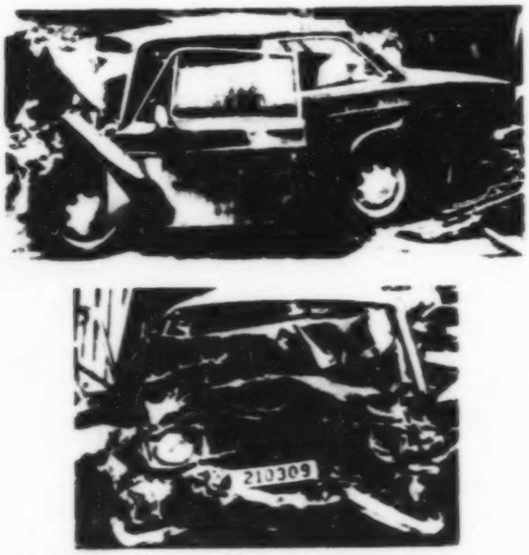
Cause of death: driver: chest injuries; rear seat passenger: head injuries.

Joint appraisal of occupant and accident vehicle.

Safety belt: probably not used.

Collision speed: above 80 km/h; estimated on extent of deformation.

VDI 12FDEW5, VIDI FS30350



Accident No. 11

ACCIDENT NO. 12

Volvo 145/1972. Type of accident: offside frontal collision with approaching truck of forward-control type (11 o'clock).

Evaluation:

1. Front seat passenger crushed between dashboard and seat backrest. Seat backrest was forced for-

ward by rear seat passenger who was not using safety belt. A six-month-old boy lying on his mother's knee was crushed to death between mother and front seat backrest.
FP 1, RP 1, Cat 1

2. Correctly used seat belts and children's seat would have had effect.
FP 1, RP 1, Cat 1
3. VESC requirements would have had effect in this accident (fairly extensive deformation of driving area, driver was belted and escaped serious injury) but would probably have had no effect for deceased.
FP 4, RP 4, Cat 2
4. Not applicable.
FP 5, RP 5, Cat 5
5. Not applicable.
FP 5, RP 5, Cat 5
6. Not applicable.
FP 5, RP 5, Cat 5
7. Not applicable.
FP 5, RP 5, Cat 5

Accident No 12

Remarks: Front seat passenger, and six-month-old boy lying on his mother's knee in rear seat, were killed. Other occupants, four persons, of which the driver was belted, escaped serious injury.

Cause of death: front seat passenger: chest injuries, rear seat passenger: head injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: (not indicated)
Collision speed: Volvo 145, approx. 60 km/h, truck 60 – 70 km/h, based on witness reports.
VDI 11FDEW6, VDI LF 20330.

ACCIDENT No. 13

Volvo 142 1968. Type of accident: frontal collision (11.30) with approaching truck of forward-control type.

Accident No 13

Evaluation:

1. Personal injuries due to contact with steering wheel.
D 1
2. Driver had no injuries other than those to abdomen and chest; therefore, safety belt would have had effect.
D 1
3. Deformation pattern is of underrun type, therefore, VESC requirements would have had no effect.
D 4
4. Not applicable.
D 5

5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5

Remarks: driver - sole occupant, killed.
Cause of death: abdominal injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: (not indicated).
Collision speed: Volvo 142, unknown; truck 70 km/h, according to driver.
VDI 12FDXA7. VIDI FS21220.

ACCIDENT NO. 14

Volvo 142/1969. Type of accident: ran off road into ditch and turned over.

Accident No. 14

Evaluation:

1. Deceased: an eight-year-old boy lay on rear parcel shelf and was ejected from car when it rolled over. Other occupants of car were only slightly injured.
   RP 4
2. Correctly worn safety belt would have had effect.
   RP 1
3. Not applicable.
   RP 5
4. Not applicable.
   RP 5
5. Not applicable.
   RP 5
6. Deformation to vehicle very slight; VESC requirements would have had no effect. Fact that rear window "popped-out," however, is of considerable importance.
   RP 4
7. Not applicable.
   RP 5

Remarks: Eight-year-old boy lying on rear parcel shelf was killed.
Cause of death: head injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: not used.
Collision speed: approx. 90 km/h, according to driver.
VDI 00TDA01. VIDI AS010000.

ACCIDENT NO. 15

Volvo 145/1971. Type of accident: head-on collision (12 o'clock) with approaching vehicle.

Accident No. 15

Evaluation:

1. Personal injuries due to contact with steering wheel.
   D 1
2. Correctly worn safety belt would have had effect.
   D 1
3. Collision speed below 50 km/h; VESC requirements would probably not have had any larger effect.
   D 3

4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Skid marks from locked wheels.
   D 2

Remarks: driver, sole occupant, killed.
Cause of death: chest and abdominal injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: probably not used.
Collision speed: Volvo 145, about 50 km/h.
VDI 12FDEW2, VIDI 0.

ACCIDENT NO. 16

Volvo 145 1969. Type of accident: skidded against tree, left side first.

Accident No. 16

Evaluation

1. Severity of accident was such that influence of interior design can be disregarded.
   D 4
2. Same as above for safety belts.
   D 4

3. Not applicable.
   D 5
4. Severity of accident considerably exceeded VESC requirements; their effect would have been limited.
   D 3
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5

Remarks: driver, sole occupant, killed.
Cause of death: head injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: not used.
Collision speed: unknown.
VDI 90LPAN5, VIDI LS 00005.

ACCIDENT NO. 17

Volvo 144 1970. Type of accident: head-on collision with pole (12 o'clock).

Accident No. 17

Evaluation

1. Personal injuries due to contact with steering wheel.
   D 1
2. Correctly worn safety belt would have had effect. Front seat passenger escaped serious injury, was using safety belt.
   D 1

3. Passenger compartment was relatively intact, VESC requirements would have had limited effect.
   D 3
4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5

Cause of death: driver: head injuries.

Joint appraisal of occupant and accident vehicle.

Safety belt not indicated.

Collision speed: 60 – 70 km/h, according to passenger and deformation extent.

VDI 12FCEN3, VIDEI S10100

Remarks

The driver, a 50-year-old woman, had her safety belt buckled. During the analysis to determine the fatal head injuries sustained, it became quite evident (from assertions by the police and the rescue people appearing first on the accident scene) that the length of belt had been very poorly adjusted. The much decreased restraint efficiency which resulted was further pronounced by the fact that the woman driver was sitting close to the steering wheel. The case investigators therefore considered it justified to conclude that the belt had not been used and the case was coded accordingly.

The front seat passenger, who had his belt properly adjusted sustained only moderate injuries.

ACCIDENT NO. 18

Volvo 142 1969. Type of accident: front loader with bucket elevated approx. 1 m. Drove out from left side in front of Volvo 142 which was overtaking. Bucket hit Volvo slightly above engine bonnet and across half the car's width.

Evaluation

1. This type of accident cannot be evaluated with regard to items 1 – 6.
   D 5
2. D 5
3. D 5
4. D 5
5. D 5
6. D 5
7. Not applicable.
   D 5

Remarks: Driver was killed and front seat passenger slightly injured.

Cause of death: head injuries.

Joint appraisal of occupant and accident vehicle.

Safety belt: not used.

Collision speed: 60 km/h, according to witnesses.

VDI 1.FEGA2, VIDEI SX0000

Accident No. 18

ACCIDENT NO. 19

Volvo 142 1970. Type of accident: ran off road and crashed into adjoining road bank.

Evaluation:

1. Personal injuries due to contact with steering wheel.
   D 1
2. Correctly worn safety belt would have had effect.
   D 1
3. Deformation localized to lower section of front; of slight extent.
   D 4
4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5

Remarks: Driver sustained injuries to chest and died 20 days later. Two passengers were slightly injured.

Cause of death: pulmonary embolism.

Joint appraisal of occupant and accident vehicle.

Safety belt: not used.
Collision speed: approx. 70 km/h, according to witnesses.
VDI 12FDLW1, VIDI 0.

Accident No. 19

ACCIDENT NO. 20

Volvo 142/1971. Type of accident: run into by rail bus on right hand side.

Evaluation:

1. Vehicle completely destroyed.
   D 4, FP 4, Car 4
2. See Item 1.
   D 4, FP 4, Car 4
3. Not applicable.
   D 5, FP 5, Car 5
4. See Item 1.
   D 5, FP 5, Car 5
5. Not applicable.
   D 5, FP 5, Car 5
6. Not applicable.
   D 5, FP 5, Car 5
7. Not applicable.
   D 5, FP 5, Car 5

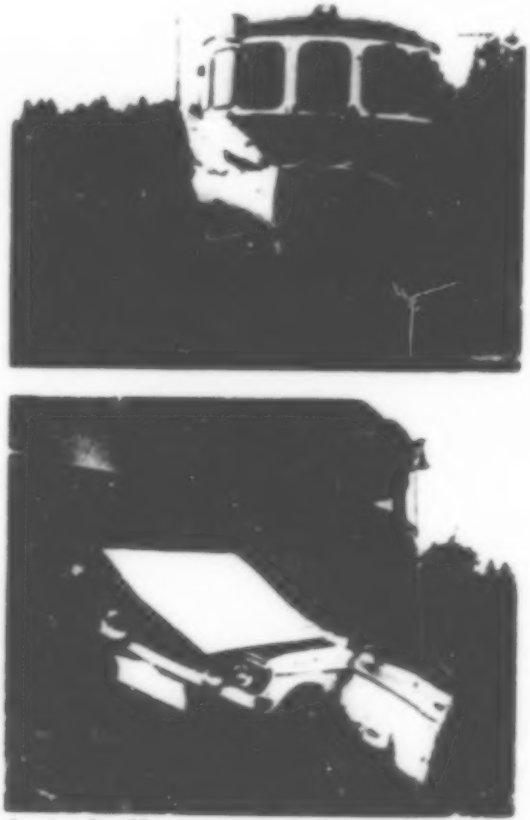
Remarks: Both the driver and front seat passenger were killed.

Cause of death: driver: total laceration; front seat passenger: total laceration.

Joint appraisal of occupant and accident vehicle:

Safety belt: (not indicated).

Collision speed: rail bus, 110 km/h, substantiated information.
VDI 03X, VIDI AS X.



Accident No. 20

ACCIDENT NO. 21

Volvo 142/1968. Type of accident: frontal collision with approaching vehicle (Volvo 145).

Evaluation:

1. Personal injuries due to contact with steering wheel.
   D 1
2. Correctly worn safety belt would have had effect. Driver of other vehicle escaped injury, was wearing safety belt.
   D 1
3. Collision speed relatively low, not above 50 km/h. Passenger compartment intact.
   D 4
4. Not applicable.
   D 5

5. Not applicable.
D 5
6. Not applicable.
D 5
7. Short skid marks, approx. 10 m.
D 4

Remarks: driver, sole occupant, killed.
Cause of death: chest and abdominal injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: not used.
Collision speed: below 50 km/h, estimated on extent of deformation.
VDI 11FDEW1. VIDI 0.

Accident No. 21

ACCIDENT NO. 22

Volvo 145/1972. Type of accident: run into by approaching car from side and rolled over.

Evaluation:

1. Design of interior would not appear to have had any importance in this accident.
RP 4
2. Correctly worn safety belt would have had effect.
RP 1
3. Not applicable.
RP 5
4. VESC requirements of no importance in this accident. Vehicle was hit behind rear wheel and spun 90° after which it turned over twice.
RP 4
5. Not applicable.
RP 5
6. Slight roof deformation.
RP 4
7. Not applicable.
RP 5

Remarks: Woman, 72 year old, sitting in front seat killed.
Cause of death: chest injuries.
Joint appraisal of occupant and accident vehicle.
Safety belt: not used.
Collision speed: Volvo 145, 70 km/h; other car, approx. 50 km/h, according to drivers.
VDI 03RBEW2 00TDAO1. VIDI RS01000.

Accident No. 22

ACCIDENT NO. 23

Volvo 145/1971. Type of accident: hit on right side by bus.

Evaluation:

1. Interior design could have been of importance.
FP 2
2. Use of safety belts would probably not have been of importance in this accident.
FP 3
3. Not applicable.
FP 5
4. Judging from deformation pattern, VESC requirements may have had effect.
FP 2
5. Not applicable.
FP 5
6. Not applicable.
FP 5
7. Not applicable.
FP 5

Remarks: Front seat passenger killed.
Cause of death: chest injuries.
Joint appraisal of occupant and accident vehicle:
Safety belt: (unknown).
Joint Appraisal of occupant and accident vehicle:
Collision speed: bus, approx. 50 km/h.
VDI 03RPAW3 : VIDERS00013

Accident No. 23

ACCIDENT NO. 24

Volvo 142/1968. Type of accident: car skidded and was rammed by approaching car on A-pillar, left side. Impact approach: 11 o'clock.

Evaluation:

1. Personal injuries can, in part, be due to contact with steering wheel.
   D 3
2. Correctly worn safety belt would probably have had some effect.
   D 3
3. Not applicable.
   D 5
4. VESC requirements for side impact would have had effect.
   D1
5. Not applicable.
   D 5
6. Not applicable.
   D 5
7. Not applicable.
   D 5

Remarks: Driver, sole occupant, killed, found after accident lying beside his vehicle.
Cause of death: chest and adbominal injuries.
Joint appraisal of occupant and accident vehicle:
Safety belt: not used.
Collision speed: Volvo 142, unknown; other car, 60 km/h, according to driver.
VDI 11LPES3 : VIDELS10001

Accident No. 24

ACCIDENT NO. 25

Volvo 142/1971. Type of accident: skidded against asphalt edge and rolled over.

Evaluation:

1. Interior design of no importance.
   D 4
2. Correctly worn seat belt would have had effect.
   D 1
3. Not applicable.
   D 5
4. Not applicable.
   D 5
5. Not applicable.
   D 5
6. Passenger compartment fully intact. Driver's head was ejected through right hand side window and was crushed between car and road.
   D 4
7. Not applicable.
   D 5

Remarks: driver, sole occupant, killed.
Cause of death: head injuries.
Joint appraisal of occupant and accident vehicle:
Safety belt: not used.
Collision speed: unknown.
VDI 00FDAO1 : VIDEO

Accident No. 25

ACCIDENT NOS. 26, 27

Volvo 145/71 and 145/72. Type of accident: head-on collision, exactly 12 o'clock and centered.

Evaluation

1. Severity of accident was such that no evaluation could be made on any of the items.
   Both 26 and 28: D4
2. Vehicle deformation 175 cm each.
   Both 26 and 27: D4
3. Possibly some effect.
   Both 26 and 27: D3
4. Not applicable.
   Both 26 and 27: D5
5. Not applicable.
   Both 26 and 27: D5
6. Not applicable.
   Both 26 and 27: D5
7. Not applicable.
   Both 26 and 27: D5

Remarks: Two drivers, each sole occupant of their cars, were killed.

Cause of death: Driver 1: chest and abdominal injuries.

Cause of death: Driver 2: chest, head and abdominal injuries.

Joint appraisal of occupants and accident vehicles

Safety belt: one driver probably wearing safety belt.

Collision speed: unknown.

VDI 12 FDAW 7 VIDLFS 41590

VDI 12 FDAW 7 VIDLFS 41590

Accidents Nos. 26, 27

ACCIDENT NO. 28

Volvo 142/1971. Type of accident: frontal collision with approaching car.

Evaluation

1. Personal injuries due to contact with steering wheel.
   D1
2. Correctly used safety belt would have had effect.
   D1
3. Collision speed below 50 km/h.
   No reduction of passenger compartment.
   D2
4. Not applicable.
   D5
5. Not applicable.
   D5
6. Not applicable.
   D5
7. Not applicable.
   D5

Accident No. 28

Accident 28 (Cont'd)

Remarks: Driver was killed, passenger escaped serious injury.
Cause of death: Driver: chest injuries.
Joint appraisal of occupant and accident vehicle
Safety belt: not used.
Collision speed (Volvo 50 km/h) estimated on extent of damage.
VDI [illegible]

ACCIDENT NO. 29

Volvo 145/1970. Type of accident: offside frontal collision with approaching car.

Evaluation

1. Severity of accident and accident situation were of such a nature that interior design, with regard to driving zone, would only have had slight effect. However, improved interior design would have been of great importance for other occupants.
   D 3, FP 2, RP 2, Car 2
2. Use of safety belt by driver would have had limited effect. For other occupants, safety belts would have had effect. Passenger compartment was relatively intact in passenger areas.
   D 3, FP 2, RP 2, Car 2
3. Collision conditions for VESC requirements were exceeded widely. Speed of Volvo 145, approx. 80 km/h and approaching vehicle approx. 100 km/h. Probability that VESC requirements on frontal collision would have been of importance for driver.
   D 3, FP 4, RP 4, Car 3
4. Not applicable.
   D 5, FP 5, RP 5, Car 5
5. Not applicable.
   D 5, FP 5, RP 5, Car 5
6. Not applicable.
   D 5, FP 5, RP 5, Car 5
7. Not applicable.
   D 5, FP 5, RP 5, Car 5

Remarks: Three of four occupants in vehicle were killed.
Cause of death: driver: head, chest and abdominal injuries; rear seat passenger: head injuries.
Joint appraisal of occupant and accident vehicle
Safety belt: unknown.
Collision speed: Volvo 145, approx. 80 km/h, other car approx. 100 km/h.
VDI 11 FDEW 7, VDI LF40492

Accident No. 29

ACCIDENT NO. 30

Volvo 142/1969. Type of accident: skidded off road, right side of car hit pole.

Evaluation

1. Passenger compartment on driver's side relatively intact.
   Interior improvements would, therefore, have had effect.
   D 2, Car 2
2. Correctly worn safety belt would have had effect.
   D 2, Car 2
3. Not applicable.
   D 5, Car 5
4. Judged on condition of vehicle after accident, VESC requirements could have been of some

importance
D 4, Car 3
5. Not applicable
D 5, Car 5
6. Not applicable
D 5, Car 5
7. Not applicable
D 5, Car 5

Remarks: driver, sole occupant, killed.
Cause of death: head injuries.

Joint appraisal of occupant and accident vehicle

Safety belt: not used.

Collision speed approx. 80 km/h, according to witnesses.

VDI 01 RYAN 3 RF 32222 LF 11002

Accident No. 30

ACCIDENT NO. 31

Volvo 145/1972. Type of accident: frontal collision with approaching vehicle.

Evaluation:

1. Front seat passenger killed. Was using safety belt but was crushed through weight of two dogs in rear seat, weighing 35 – 40 kg each. Driver without safety belt. Moderate injuries.
FP 1
2. Correctly worn safety belts would have had effect.
FP 1
3. VESC requirements of no importance. Speed below or about 50 km/h. No reduction of passenger compartment.
FP 4
4. Not applicable
FP 5
5. Not applicable
FP 5
6. Not applicable
FP 5
7. Not applicable
FP 5

Remarks: Front seat passenger, a woman wearing a safety belt died after approximately 30 days, due to changes in condition of lungs resulting from accident.

Joint appraisal of occupant and accident vehicle

Safety belt: used by front seat passenger but not by driver.

Collision speed: approx. 50 km/h.

VDI 12 FDEW 2 VDI 0

Accident No. 31

THE SWEDISH TECHNICAL PRESENTATION

SECTION **TWO** PART **6**

ROAD ACCIDENT INVESTIGATION—ACCIDENTS IN SWEDEN WITH SAAB 99, REPORT FROM FIRST PHASE

HENRIK GUSTAVSSON, *Director Engineering*
Saab-Scania

At the beginning of 1971, Saab-Scania decided to conduct a road accident investigation as a further aid to developing safer cars. It would in essence follow the CCMS Pilot Study on Road Safety.

This paper presents the evaluation of data collected during the first phase of the investigation, from May 1, 1971 to April 30, 1972 and from September 1, 1972 to November 30, 1972. The total number of accidents covered was 1,525, of which 158 were in-depth investigations.

When planning the investigation we decided to take advantage of the fact that all later model Saab cars sold on the Swedish market have a five year vehicle-damage insurance included in the basic sales price. All damage above a cost of 500 Sw.Cr. ($110 U.S.) is paid by this insurance and all cars damaged above 7,000 Sw.Cr. ($1,550 U.S.) are replaced. The damaged vehicle is sent to a Saab-Scania-owned central repair workshop.

It was decided that only Saab cars of the "99" model, that were replaced and sent to the central workshop, would be included. To avoid excessive delays due to long travel distances, it also was decided that the in-depth investigation should cover only that part of Sweden which is south of a line between the towns of Gävle and Falun. About 85 percent of the population live in this southern area, so the majority of the accidents would be investigated (Figure 1).

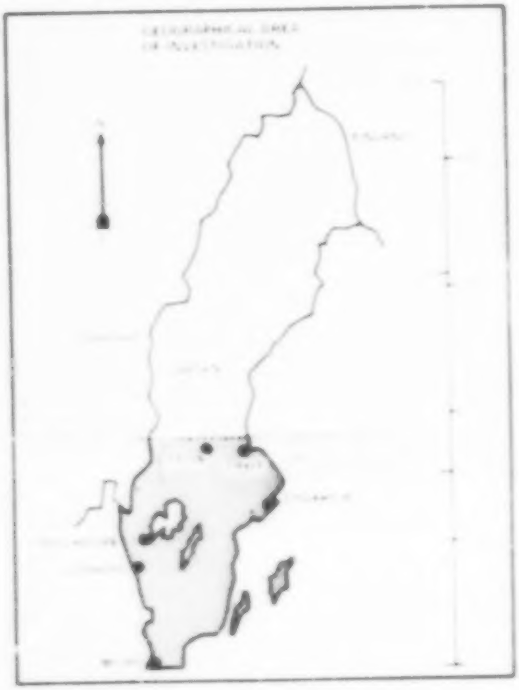


Figure 1

The "in-depth investigation" was divided into three zones:

1. The area around the towns of Trollhättan, Vänersborg, and Uddevalla which are close to the main plant, where investigations on the accident site could be carried out.
2. The area where our dealer workshops could be reached in half a day. No accident-site investigations were conducted, vehicles were inspected at the dealers.
3. Area not covered by 1 and 2: no accident-site investigation. Vehicles were inspected at the central repair workshop in Nyköping.

Information about occupant injury was collected from the various hospitals involved, occupants were interviewed and vehicles examined. When all obtainable data regarding an accident were present, the final evaluation was carried out by medical and technical experts. This evaluation emphasized the *cause of injury* of all 168 accidents fitting the above criteria. Ninety-four percent were investigated (158), and none of the 10 that slipped had fatal or severe injuries.

To ascertain whether severe injuries occurred in accidents with vehicle damage below the earlier mentioned 7,000 Sw.Cr. ($1,550 U.S.) a separate "survey" was performed. This survey covered *all accidents* with Saab 99's in Sweden during September, October and November 1972 with vehicle damage above 500 Sw.Cr. ($110 U.S.). When an accident with vehicle damage was reported, the report

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form included questions about occupant injury and seat belt usage. If an injury was reported, a short interview was held with the occupant and in some cases information was collected from hospital files.

The road road-accident investigation is now in its second year and includes both in-depth investigations of severe accidents and a survey of lighter accidents.

ACCIDENT DISTRIBUTION VERSUS REPAIR COSTS

The diagram in Figure 2 covers 1,156 accidents from the "survey period" which were all accidents with vehicle damage above $110 U.S., south of the line shown on the map in Figure 1 as the in-depth investigation area.

Ninety-six percent of all accidents had vehicle damage between $110 and $1,550 U.S. and six percent had more severe damage. We can thus see that our "in-depth investigation" covers about six percent of all accidents above liability.

VEHICLE MODEL YEAR

The distribution on vehicle model year in the in-depth investigation year was M-69, 28 percent, M-70, 32 percent, M-71, 34 percent and M-72 6 percent of all 158 cases (Figure 3). These figures reflect the number of different model year vehicles on the road.

The one-year investigation period started May 1, 1971, which means that the majority of M-71 vehicles was exposed during the whole investigation period, but the M-72 exposure varied from a maximum of nine months down to zero.

The vehicles were basically the same over these different model years with only smaller changes, like the M-70 with new interior-door panel and arm rests, the M-71 with a new instrument panel, and the M-72 with energy-absorbing bumpers.

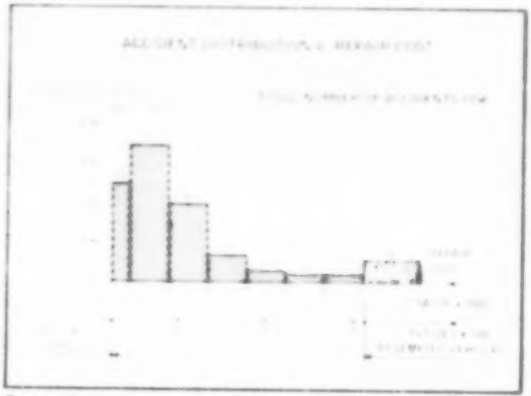


Figure 2

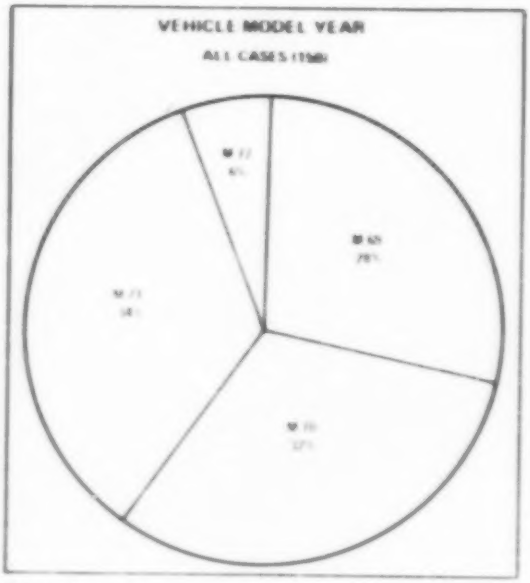


Figure 3

AREA

Seventy percent of all accidents in the in-depth investigation (158) were rural and 30 percent urban (Figure 4).

The frequency of occupant injury in rural accidents was 58 percent, compared with 72 percent in urban accidents. This was probably due to more accidents of the single-vehicle type happening on

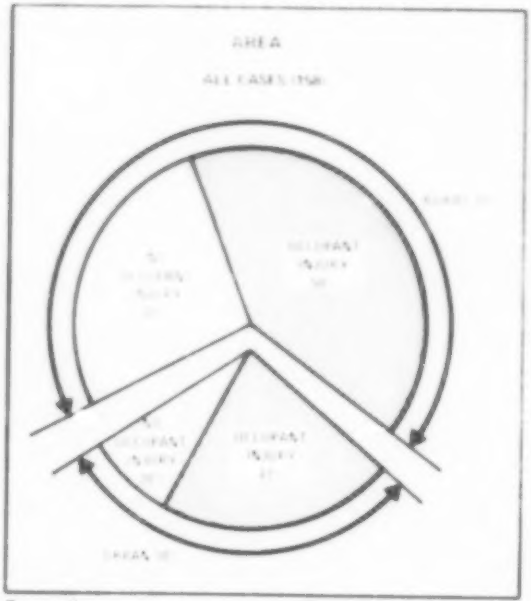


Figure 4

country roads where vehicles can run off the road without being subjected to too high decelerations.

SPEED LIMIT

The diagram in Figure 5 shows the accident distribution and occupant injuries versus speed limits for the in-depth investigation (158 cases). Fifty km/h is the common speed limit in urban areas. 70 km/h is the basic limit on rural roads. 90 km/h can be allowed by local authorities if traffic density is less than 7,000 vehicles per day and the road is more than 5-meters wide. On motor-ways and some other very good roads, 110 km/h is allowed.

No clear indication can be found that lower speed limits reduce the frequency of occupant injury. (For injury severity see Figure 12 on page 397.)

TYPE OF ACCIDENT

From In-depth Investigations of 158 Cases (Figure 6)

Vehicle-to-vehicle accidents were 56 percent and vehicle-to-other objects, 44 percent. As could be expected, frequency of occupant injury was higher in vehicle-to-vehicle accidents, 72 percent versus 51 percent, and is mainly explained by the difference in vehicle deceleration.

IMPACTED OBJECTS

From In-depth Investigations of 158 Cases (Figure 7)

The most common impacted objects were other passenger cars, with 35 percent of all accidents. Impacts against trucks were 11 percent, but the frequency of occupant injury was higher, 80 percent, compared with 68 percent when another passenger car was impacted. Occupant injury also was much more severe in impacts against trucks.

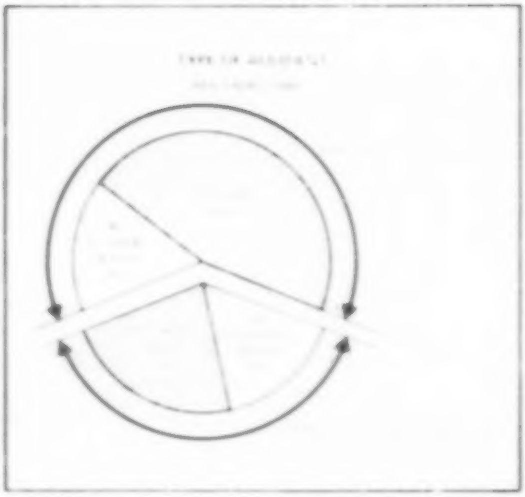


Figure 6

TYPE OF COLLISION

From In-depth Investigations of 158 Cases (Figure 8)

Frontal collisions and other collisions had a frequency of occupant injury of 67 versus 63 percent. Rear-end collisions seemed to be less dangerous at 50 percent (about 1/3 of the vehicles had head restraints) and side collisions more dangerous at 73 percent frequency of occupant injury.

DIRECTION OF IMPACT

From In-depth Investigations of 158 Cases (Figure 9)

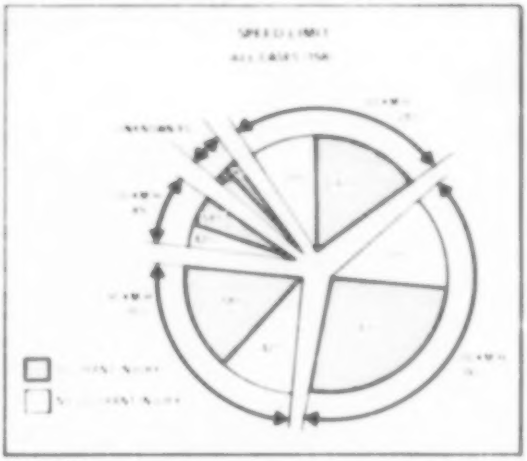


Figure 5

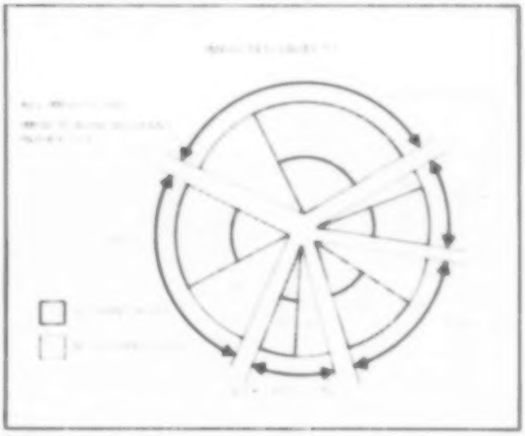


Figure 7

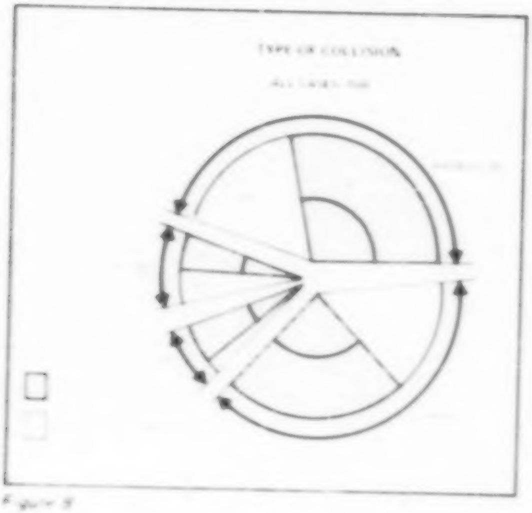


Figure 8

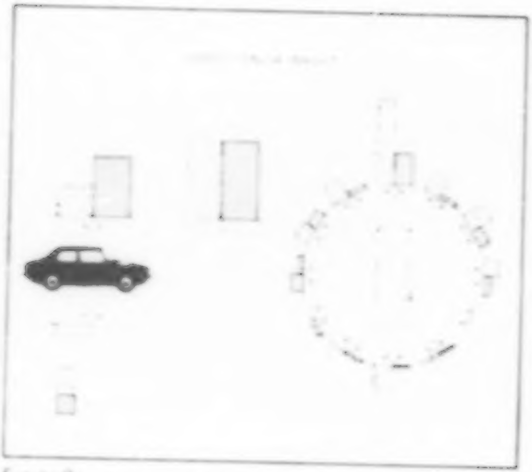


Figure 9

INTERIOR DEFORMATION, FRONTAL

From In-depth Investigations of 158 Cases (Figure 10)

The distribution of vehicle-interior deformation index (VIDI) shows that 83 percent of all vehicles involved had no such deformation.

INTERIOR DEFORMATION, SIDEWAYS

Front In-depth Investigations of 158 Cases (Figure 11)

The distribution of VIDI shows that 57 percent of all vehicles involved had no interior side deformation and 32 percent had a VIDI of 1. (Compare Figures 7, 9 and 10.)

OCCUPANT INJURY VS. SPEED LIMIT

From In-depth Investigations of 146 Occupants, 5 of Which Are Unknown (Figure 12)

The severity of occupant injury increases with speed. The average injury index is slightly higher at 50 km/h than at 70 km/h, which is probably due to a larger part of side collisions in urban areas. At 110 km/h the number of injured occupants was only nine and the average index thus uncertain.

OCCUPANT INJURY IN RELATION TO DIRECTION OF IMPACT AND VEHICLE DEFORMATION INDEX

From In-depth Investigations (Figure 13)

Impact directions from 9 and 10 o'clock seemed to cause occupant injury at a lower vehicle deformation index than at other directions. But Figure 8 also shows a larger risk of occupant injury in side

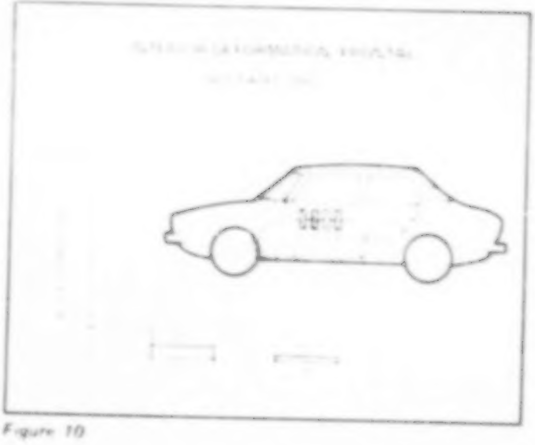


Figure 10

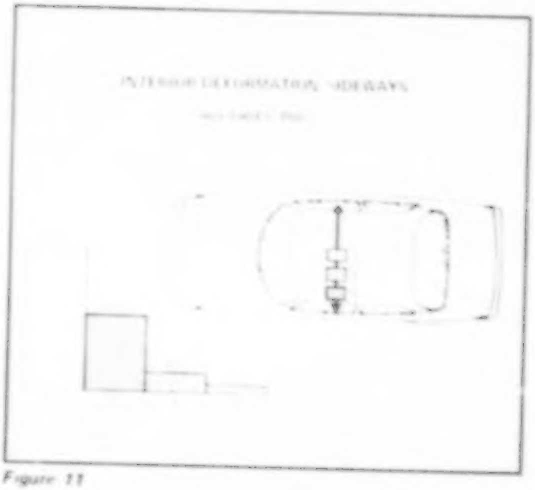


Figure 11

OCCUPANT INJURY V. SPEED LIMIT

SPEED LIMIT		NO. OF INJURED OCCUPANTS	INJURY INDEX AVERAGE
KPH	MPH		
50	31	33	1.67
70	44	70	1.64
90	56	29	2.24
110	68	9	2.00

TOTAL NUMBER OF OCCUPANTS 146
(UNKNOWN 5)

Figure 12

collision. The driver's seat is, of course, occupied more frequently than passenger seats.

CAUSE OF INJURY RELATED TO INJURY INDEX AND SEAT BELT USAGE

From In-depth Investigations of 158 Accidents (Figure 14)

The relatively high placed instrument panel does not always prevent the occupant from hitting the firewall. This area is very stiff and attempts to add protection in the form of insulation and energy-absorbing panels but, as yet, cannot be regarded as successful. Nine injuries with an index of 3-9 were recorded. Other areas that caused injuries with an index of 3 or higher were, for example, the steering wheel, front door panel and instrument panel.

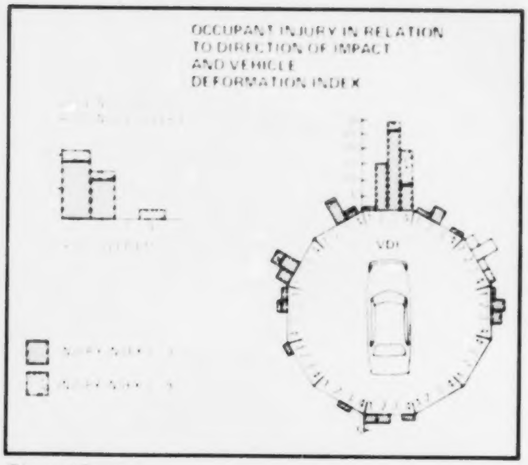


Figure 13

CAUSE OF INJURY RELATED TO INJURY INDEX (Low Frequency and Unspecified Causes)

From In-depth Investigations of 158 Accidents (Figure 15)

CAUSE OF INJURY RELATED TO INTERIOR MODIFICATIONS AND INJURY INDEX

From In-depth Investigations of 158 Accidents (Figure 16)

The introduction of a new instrument panel in the M-71 seems to have reduced the severity of injuries caused by impacts. The panels that were attached to this new instrument panel to reduce injuries from impacts into the fire wall were not effective enough.

SEAT BELT USAGE

From In-depth Investigations and Survey Period; 1,523 Accidents (Figure 17)

The usage of seat belts at 29 and 36 percent, respectively, was higher than we anticipated, especially the higher figure. The seven percent increase between the in-depth investigations and the survey period could have two different causes. (The in-depth investigations ended in the spring of 1972.) In early summer, a seat belt campaign was launched in Sweden, after the in-depth investigation, but before the start of the survey period. In addition, we feel that the people interviewed by telephone might tend to exaggerate the truth.

CAUSE OF INJURY

Related to Injury Index 1-9 and Seat Belt Usage

DISTRIBUTION		INJURY INDEX									INDEX 3-9	GRAND TOTAL
CAUSE		1	2	3	4	5	6	7	8	9	TOTAL	
Steering wheel		16	2	1	1		1				3	25
Steering wheel	belted	4									0	
Front door panel		9		2							2	20
Front door panel	belted	6	2	1							1	
Seat belt	belted	12	2		2						2	16
Firewall		2		7	1						8	12
Firewall	belted	1		1							1	
Lower instr. panel left		7	1								0	12
Lower instr. panel	belted	4									0	
Windshield right		6	5								0	11
Instrument panel right		3	1		1	1			1		3	9
Instrument panel	belted	2									0	
Front seat lower frame		9									0	9
Other occupant		1	5	1							1	8
Other occupant	belted				1						1	
Seatback		4	2								0	7
Seatback	belted	1									0	
Rearview mirror		7									0	7
Flying glass particles		3									0	6
Flying glass particles	belted	3									0	
Objects outside of vehicle				1		1	1			2	5	5
A post		4				1					1	5
Front side window frame		2	2								0	5
Fr. side window frame	belted	1									0	
Header		3	2								0	5
Windshield left		4	1								0	5

Note: Occupants belted only where expressly stated.
Total number of accidents 158.

Figure 14

CAUSE OF INJURY

Related to Injury Index 1-9
Low Frequency and Unspecified Causes

DISTRIBUTION		INJURY INDEX									INDEX 3-9 TOTAL	GRAND TOTAL
CAUSE		**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**		
Side header		3									0	4
Side header	belted		1								0	
Grab handle		3									0	3
Gearchange lever		3									0	3
Pedals		3									0	3
Whiplash		3									0	3
Ejection through door		3									0	3
Lower instr. panel right		1	1								0	2
Radio		1	1								0	2
Roof		2									0	2
Rear side window frame		2									0	2
Exterior of vehicle		2									0	2
Rear side window						1					1	1
Rear side interior panel				1							1	1
Ashtray			1								0	1
B post			1								0	1
Rear window		1									0	1
Front side window		1									0	1
Glasses		1									0	1
Other		4	1								0	8
Other	belted	2	1									
Unknown		38	9	2			1				3	63
Unknown	belted	12	1								0	

Note: Occupants belted only where expressly stated.
Total number of accidents 158.

Figure 15

CAUSE OF INJURY

Related to Interior Modifications and Injury Index 1-9

DISTRIBUTION		INJURY INDEX									INDEX
CAUSE	MODEL	1	2	3	4	5	6	7	8	9	3-9 TOTAL
Front door panel	70 72	5	1	1							1
Front door panel	69	10	1	2							2
Firewall	71 72			3	1						4
Firewall	69 70	3		5							5
Lower instr. panel left	71 72	7									0
Lower instr. panel left	69 70	4	1								0
Instrument panel right	71 72	3	1								0
Instrument panel right	69 70	2									0
Grab handle (new)	71 72	3									0
Lower instr. panel right	69 70	1	1								0
Radio	71 72										0
Radio	69 70	1	1								0
Ashtray	71 72		1								0
Ashtray	69 70										0

Total number of accidents 158

Figure 16

SEAT BELT USAGE

	OCCUPANTS		
	TOTAL	BELTED	PERCENTAGE BELTED
IN DEPTH INVESTIGATION	283	82	29
SURVEY PERIOD	2065	744	36
TOTAL	2348	826	35
TOTAL NUMBER OF ACCIDENTS	1523		

Figure 17

OCCUPANT INJURY V. CRASH SEVERITY AND SEAT BELT USAGE

From In-depth Investigations and Survey Period; 1,523 Accidents (Figure 18)

The frequency of occupant injury in severe accidents was reduced from 57 percent without seat belts to 34 percent with seat belts. Regarding injury frequency in less severe accidents also refer to comments made on the survey in Figure 17.

INJURED OCCUPANTS WITH AND WITHOUT SEAT BELT

From In-depth Investigations of 158 Accidents (Figure 19)

The frequency of occupant injury in the front seat was reduced from 61 percent without seat belts to 38 percent with seat belts. Only one rear seat passenger used the seat belt; the figures for rear seat passengers are shown in a separate column.

OCCUPANT INJURY V. CRASH SEVERITY AND SEAT BELT USAGE

SEVERITY OCCUPANTS	REPAIR COST BELOW $1550	7000 SW.CR	REPAIR COST OVER $1550	700 SW. CR
WITH SEAT BELT	689		137	
INJURED	14	(2.0%)	46	(34%)
WITHOUT SEAT BELT	1217		285	
INJURED	20	(1.6%)	162	(57%)
ALL OCCUPANTS	1906		422	
	34	(1.8%)	208	(49%)
TOTAL NUMBER OF ACCIDENTS	1523			

Figure 18

INJURED OCCUPANTS WITH AND WITHOUT SEAT BELTS

	DRIVER	FRONT PASS.	TOTAL	REAR PASS.
WITH SEAT BELT	56	25	81	1
INJURED	20 (36%)	11 (44%)	31 (38%)	0
WITHOUT SEAT BELT	102	39	141	60
INJURED	57 (56%)	29 (74%)	86 (61%)	29 (48%)

Figure 19

TOTAL INJURIES WITH AND WITHOUT SEAT BELT

From In-depth Investigations of 242 Injuries (Figure 20)

The frequency of injuries of Index 3-9 was more than halved, from 14 percent to 6 percent when seat belts were used.

Note that each occupant can have more than one injury.

INJURY LOCALIZATION, DRIVER

From In-depth Investigations of 77 Injured Drivers, (Figure 21)

The frequency of head injuries was reduced from 30 percent without seat belts to 15 percent with seat belts. Torso and upper limb injuries of minor character increased when seat belts were used.

Injuries are distributed on following broad areas:

Head and neck
Upper limbs
Torso
Lower limbs

INJURY LOCALIZATION, FRONT PASSENGER

From In-depth Investigations of 40 Injured Passengers (Figure 22)

The frequency of head injuries was reduced from 42 percent without seat belts to 16 percent with seat belts. Torso injuries of a minor character also increased with seat belts.

INJURY LOCALIZATION, REAR PASSENGER

From In-depth Investigations of 29 Passengers Without Seat Belts (Figure 23)

TOTAL INJURIES WITH AND WITHOUT SEAT BELT

	INDEX	DRIVER	FRONT PASS.	TOTAL	
WITH SEAT BELT	1 2	46	18	64	(94%)
	3 9	1	3	4	(6%)
WITHOUT SEAT BELT	1 2	99	50	149	(86%)
	3 9	16	9	25	(14%)

Figure 20

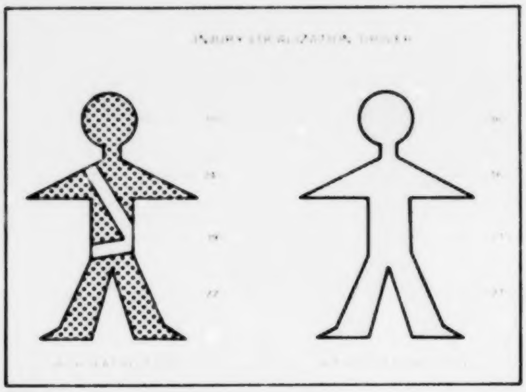


Figure 21

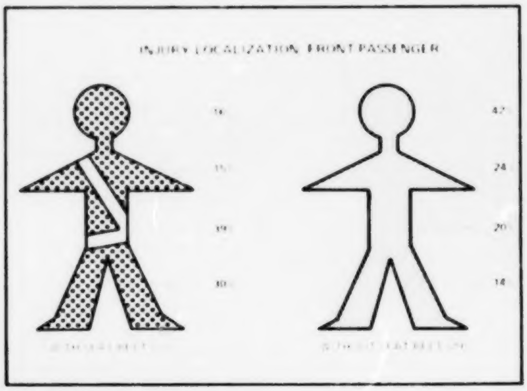


Figure 22

DISTRIBUTION OF OCCUPANT INJURY ACCORDING TO INJURY INDEX

From a Survey of 1,284 Accidents With Damage Below $1,550 U.S. (Figure 24)

From In-depth Investigations of 158 Accidents (Figure 25)

Of the accidents in the survey that had vehicle damage between $110 to $1,550 U.S., 96.4 percent caused occupant injuries of Index 1 or 2. No injuries with Index over 4 were found.

Of the accidents in the in-depth investigations 89 percent caused injuries of Index 1 or 2.

Three percent of the injuries had an Index over 4.

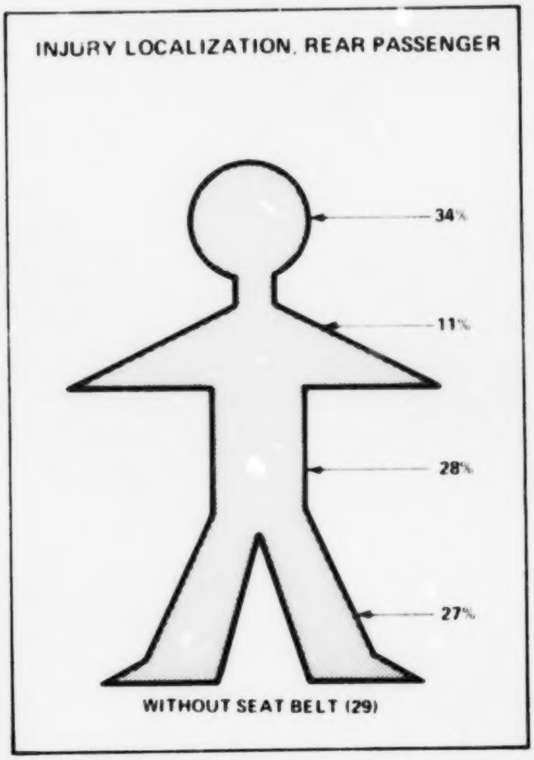


Figure 23

TREATMENT

From In-depth Investigations of 284 Occupants (Figure 26)

Fifty-one percent of the occupants did not visit a doctor.

Eighteen percent of the occupants were hospitalized.

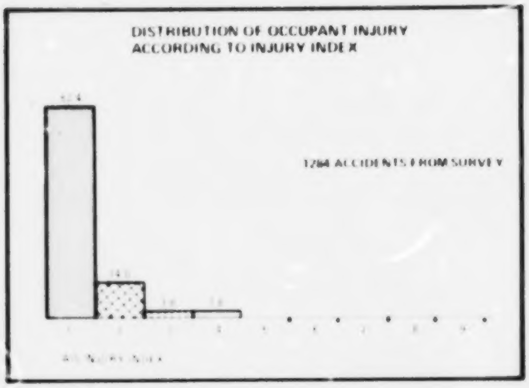


Figure 24

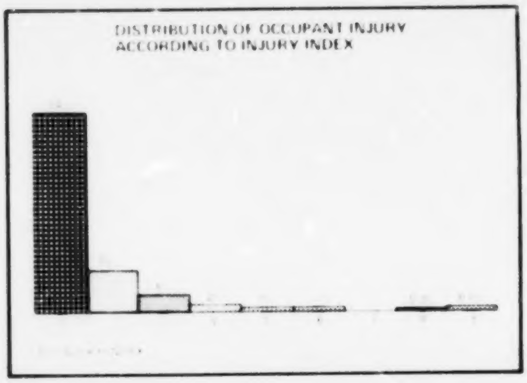


Figure 25

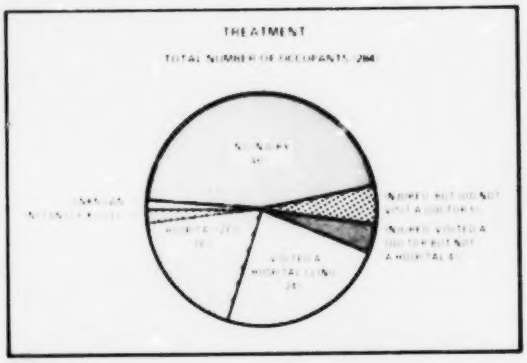


Figure 26

TIME OF REHABILITATION V. OCCUPANT INJURY INDEX

From In-depth Investigations of 117 Occupants (Figure 27)

The number of occupants that received injuries of Index 4 or 5 was too small to give reasonably certain figures.

Our road-accident investigation is now about halfway through its second year, which will end in August 1973. All data gathered from both periods will be computerized to give easy egress to any combination of data. We are still putting our main effort into detailed investigations of severe accidents and believe that this way we gain more knowledge in less time, knowledge that will be used when new and safer designs are made.

TIME OF REHABILITATION

V. OCCUPANT–INJURY INDEX

INJURY INDEX	UNFIT FOR WORK, DAYS TOTAL	NUMBER OF INJURED OCCUPANTS	AVERAGE TIME UNFIT FOR WORK, DAYS
1	765	83	9
2	841	25	34
3	1141	9	127

NOTE TOTAL TIME INCLUDES SATURDAYS, SUNDAYS ETC. UNFIT FOR WORK

Figure 27

VEHICLE EXTERIOR DAMAGE

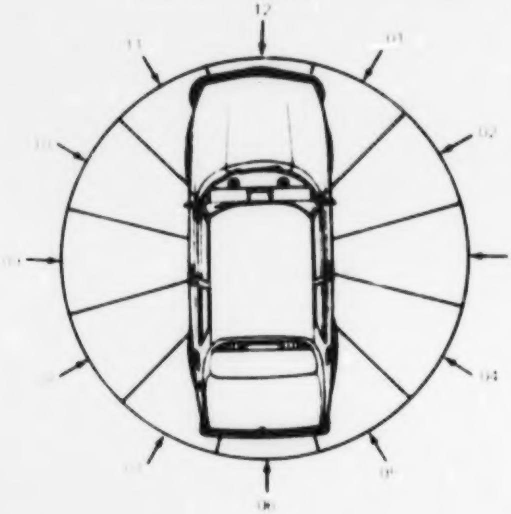
TYPE OF COLLISION

Main definitions:

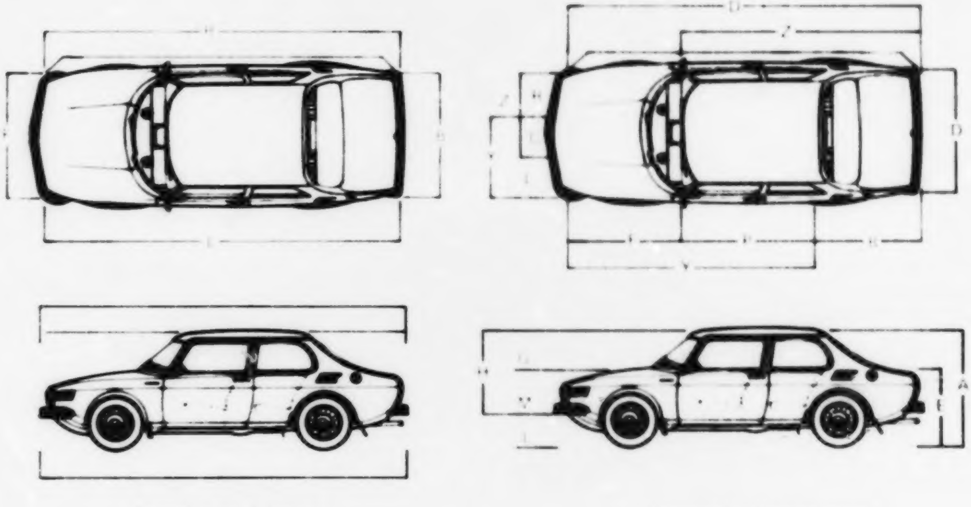
Wide object
Narrow object
Side impact
Overturning
Underride
Edge impact
Fire
Explosion and fire
Water or hazards
Unknown

DIRECTION OF IMPACT

Illustrated in clock diagram



LOCATION OF IMPACT



General zones of damage

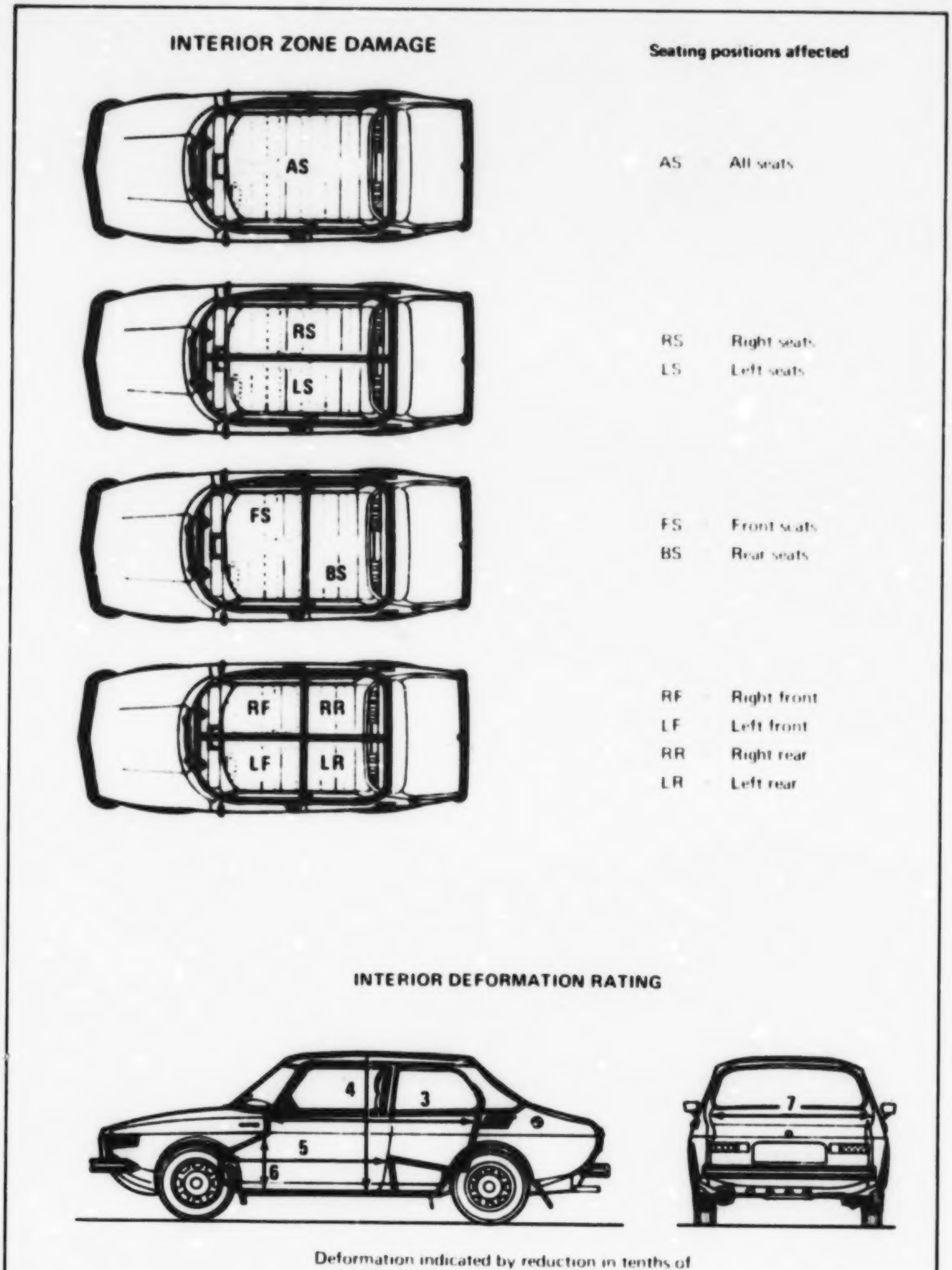
Front, right, rear, left top and underbody main zones.

Horizontal and vertical zones

Damage in detail zones as illustrated.

Deformation indicated by reduction in tenths of original dimensions.

Figure 1


INTERIOR ZONE DAMAGE
Seating positions affected
AS
RS
LS
FS
BS
RF
RR
LF
LR
INTERIOR DEFORMATION RATING
4
3
5
6
7

AS	All seats
RS	Right seats
LS	Left seats
FS	Front seats
BS	Rear seats
RF	Right front
LF	Left front
RR	Right rear
LR	Left rear

Deformation indicated by reduction in tenths of original dimension

Figure 2

OCCUPANT INJURY INDEX (AIS)

As Per American Medical Association

Injury Index	Severity Category	Injury Description, Examples
0	No injury	Minor bruises.
1	Minor	Muscle ache. Minor lacerations, contusions and abrasions. Teeth fracture. Minor sprains. Sign of cerebral injury without loss of consciousness.
2	Moderate	Extensive contusions, abrasions and large lacerations. Cerebral injury with or without skull fracture. Severe "whiplash" neck pain with evidence. Simple rib fracture. Undisplaced long bone fracture.
3	Severe, not life threatening	Cerebral injury with more than 15 min unconciousness. Loss of eye. Moderate spine fracture. Multiple rib fractures, lung contusion. Displaced long bone or pelvic fractures. Multiple hand and foot fractures.
4	Serious, survival possible	Severe lacerations, dangerous hemorrhage. Compound skull fracture. Open chest wounds. Ruptured spleen or kidney. Amputation of limbs.
5	Critical, survival uncertain	Over 50 % 2 or 3 burns. Cerebral injury with over 24 hours unconciousness. Chest injuries with breathing difficulty. Aortic laceration. Severe injury of intra-abdominal organs.
6-7	Fatal within 24 hours	Fatal lesions of single body region plus non-fatal injuries of other regions.
8-9	Fatal	Two or more fatal injuries.

Figure 3

THE ITALIAN TECHNICAL PRESENTATION

SECTION TWO

PART 7

INTRODUCTION

PROFESSOR AUGUSTO SIRIGNANO, *Director*
General for Motorization
Ministry of Transport

The Italian delegation has the pleasure of introducing Engineer Moscarini of the Experimental Institute on Motors who will give an account of the research carried out in the "Field of Vibrations by Air and by Solid Bodies on the Human Organism."

We should like to specify that the Experimental Institute, represented here by Mr. Moscarini, is a private institute, and a research study experimental institute, which operates independently from any other industrial organization

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THE ITALIAN TECHNICAL PRESENTATION

SECTION **TWO** PART **7**

PROGRESS REPORT FOR THE EXPERIMENTAL INSTITUTE FOR MOTOR VEHICLES (ISAM)

FILIPPO MOSCARINI, *Director*
Institute for Experiments on
Automobiles and Motors
Alfa Romeo

EFFECT OF VIBRATIONS BY AIR AND BY SOLID BODIES ON THE HUMAN ORGANISM

INTRODUCTION

A large number of technical experts agree that drivers are responsible for 90% of road accidents. Taking this into account, the Experimental Institute for Motor Vehicles has begun a research study on the vibrations found inside motor vehicles. This study is based on a vast bibliography, of which we have considered only the part dealing with low-frequency sound and mechanical vibrations, viz. those from 2 to 100 Hz. Obviously, this research is the hardest because of the difficulty of observing low frequencies; which, however, appear to be the most dangerous, since they do not cause immediate unpleasant sensations, and are not readily noticed, although they adversely affect both perception and the cardiovascular system.

In Italy, these studies were first made by the Naval Research Institute on transatlantic liners. It frequently happened that in the large lounges of the liners, when the sea was rough, the walls undulated with quite considerable deflection, creating sound waves of very high and low frequency, which caused conditions of serious malaise among the passengers. Because of the difference in area affected by the phenomenon we cannot, of course, expect to find similar intensities in motor vehicles. However, in our case, the problem takes on another aspect, namely that of the time during which the passenger is subjected to these vibrations.

Research carried out so far seems to confirm that the concept of the sum of effects in time holds good; and, taking into consideration the average driver, it can be seen that in the course of several years, the number of hours during which he is subjected to low-frequency vibrations, amounts to several thousands.

In this connection, ISO published in 1972, a guide assessing the exposure of the human body to vibrations, in which various physiological aspects of the human body are related to the intensity, frequency and duration of the vibrations.

When we found that the effects can be of various kinds, and can cause a considerable reduction in "active safety" which, on the contrary, should be maximized, it was thought that it might be helpful to investigate the presence or absence of these phenomena in a motor vehicle.

We found in a large part of our bibliography, that people subjected for a time to vibrations of a certain intensity and frequency, suffer a reduction of vision and are in a state of trance or, worse still, in a state of euphoria similar to that brought about by a generous consumption of alcohol.

Accordingly, after weighing these facts, and bearing in mind that the driver is the person most responsible for road accidents, ISAM thought it necessary to carry out a careful investigation to determine the amplitude and intensity of the vibrations found in a motor vehicle.

OPERATION SYSTEM

The investigation in question was carried out by ascertaining the presence and entity of the aforesaid phenomena on a specimen car at the first stage; in the second stage, the most significant tests for the observation were repeated on a second vehicle. The course of the study, then, envisaged an ample and general statistical formulation.

Three situations were analyzed for the specimen vehicle:

1) On a motorway, where observation was made of infrasounds, low-frequency sounds and vibrations of 2-100 Hz frequency during the gradual passage in fourth gear, with low and steady acceleration, from 1,000 revolutions per minute to 4,000 revolutions per minute.
2) With the vehicle stationary and the engine in neutral, where observation was made during the gradual passage from 1,000 revolutions per minute to 5,000 revolutions per minute.

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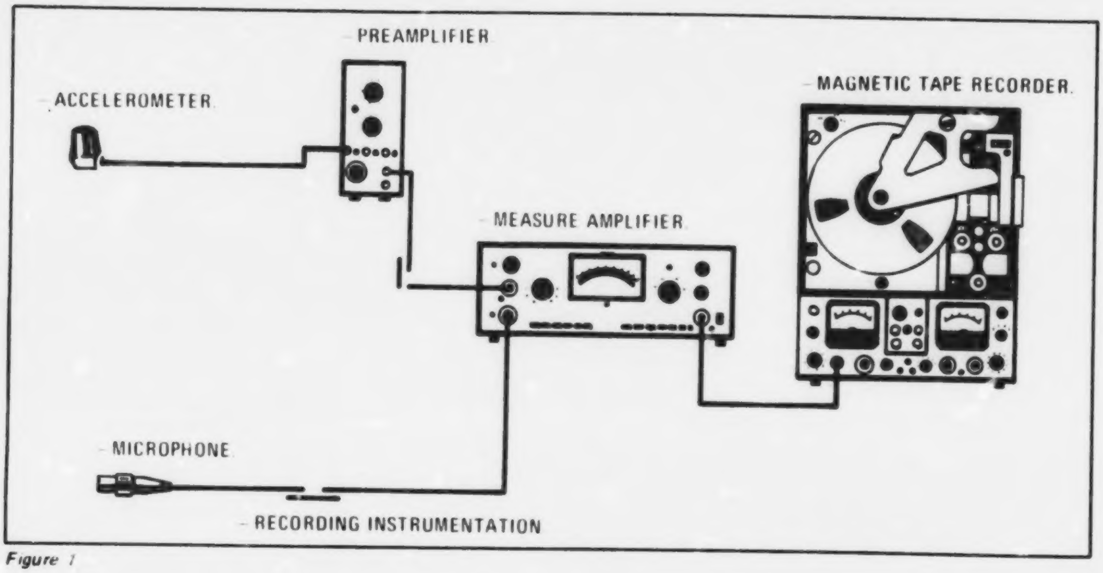

Figure 1

3) With the car placed (with the driving wheels resting) on a roller test bench with inertial masses, where observation was made during the gradual passage in fourth gear from 1,000 revolutions per minute to 4,000 revolutions per minute.

These situations were chosen for the first analysis of the problem for the following reasons:

The first situation was chosen because it provided information on the general behaviour of a motor vehicle in motion along a road. Moreover, since low and steady acceleration was chosen, each individual speed could be considered an actual running speed.

The second situation was chosen because it enabled us to isolate the engine from the rest of the structure as responsible for part of the infrasounds, sounds and vibrations present.

The third situation was chosen to obtain information on critical conditions for the vehicle. But it was actually found that this situation has little significance, since the motor vehicle must have the pair of non-driving wheels blocked, as in the natural position on a roller test bench.

When these typical conditions for the car had been chosen, observation of the relevant phenomena was carried out by first recording them on magnetic tape, and then analyzing and recording them on paper. In other words, the procedure was carried out in accordance with two points:

A) Observation and recording on tape
B) Analysis and recording on paper.

Point A: Observation and Recording on Tape

The arrangement of the instruments used at the observation stage is shown in Figure 1.

Figure 2 shows the arrangement of the same instrument on the back seat of the vehicle.

A condenser microphone was used as a sensory transducer of infrasounds and low-frequency sounds. This was attached to the rearview mirror of the motor vehicle, as shown in Figure 3.

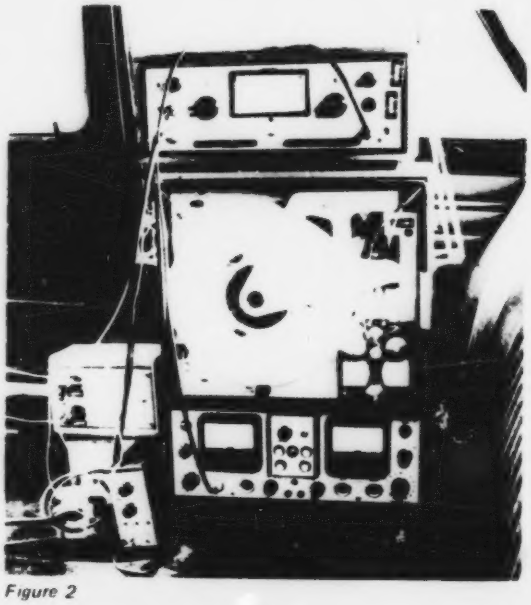

Figure 2

Figure 3

The signal transmitted is sent to a measuring amplifier and then to a tape recorder. A recording speed of 1.5 ins. per sec. was chosen for the tape recorder. This speed is responsible for the upper limit of the frequencies that can be recorded; this limit is found to be equal to 500 Hz.

A similar procedure was adopted for observing the 2-100 Hz frequency vibrations. However, a piezoelectric accelerometer was used as sensory transducer. The accelerometer was attached magnetically to the steering wheel of the front seat, and the observation was made by facing it in all three directions, viz. longitudinally to the driving direction, and transversally both in a vertical and horizontal sense. Figure 4 shows the position of the accelerometer.

Point B: Analysis and Recording

After the relevant signals in the three situations described above were observed and recorded, an analysis was made which took into account both the spectrum of the frequencies and the amplitudes of the signals. In order to obtain the frequency spectrum, the following method was adopted:

The speed of the tape recorder was increased tenfold, viz. from 1.5 ins. per sec. to 15 ins. per sec.; thus, a transposition of frequency was obtained. This was done because no analyzers are available for low frequencies, such as those in which we are interested. Consequently, every signal recorded during the individual stages of observation was sent to a digital recorder and accordingly memorized.

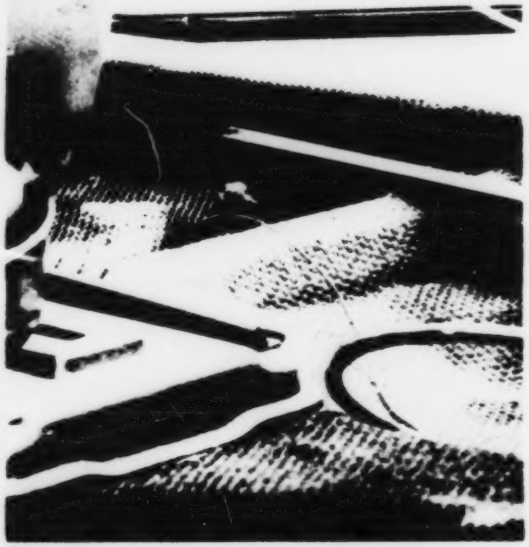

Figure 4

Next, a harmonic analysis was made of the comprehensive signal for the duration of each individual stage of observation by means of an analyser in the actual time at 1/3 of an octave.

The arrangement of the instruments used at this stage can be seen in Figure 5.

To sum up, then, for transient phenomena the digital recorder memorizes the signal encountered on the tape recorder; this signal is then analyzed in frequency by the analyzer; the frequency spectrum is moreover made visual on a screen, and then recorded on paper. Everything is much easier for the analysis and recording on paper of the amplitude of the signals observed, for one only has to transpose onto paper the recordings on the magnetic tape.

The following graphs show the amplitudes of the signals observed in relation to the number of the engine's revolutions.

Thus, Figure 6 gives a diagram of the spectrum of vibrations by air between 2 and 500 Hz, recorded on the vehicle in the three different working conditions, viz. on the road, on the rollers and in neutral.

Figure 7, on the other hand, shows the diagram of the spectrum of vibrations by solid bodies along the three axes. This was recorded on the same vehicle as in the preceding diagram, in the two working conditions, viz. on the road and in neutral.

Figure 8 shows the diagram of the spectrum of the vibrations between 2 and 500 Hz by solid bodies: again along the three axes, in two different vehicles in the same running conditions on the road.

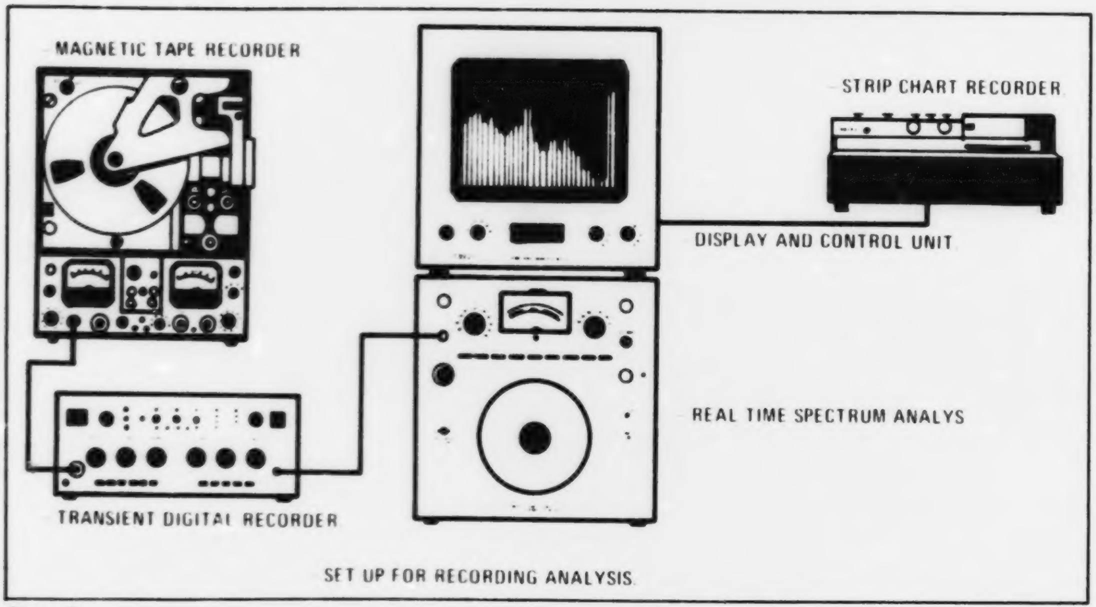
MAGNETIC TAPE RECORDER
STRIP CHART RECORDER
DISPLAY AND CONTROL UNIT
REAL TIME SPECTRUM ANALYS
TRANSIENT DIGITAL RECORDER
SET UP FOR RECORDING ANALYSIS

Figure 5

We now have both the amplitudes and the frequency spectrum of the signals and can proceed to a first analysis of the results.

It can immediately be noted that in the field of frequencies considered by us, there are levels high enough to affect the well-being of the human organism; that is to say, levels so high as to cause a marked reduction in "active safety."

An indication of the values of the levels high enough to jeopardize the health of the human organism has been derived from the ISO survey already quoted. As we have seen, this report also takes into consideration the time of exposure to the phenomena. In any case, since the presence of already

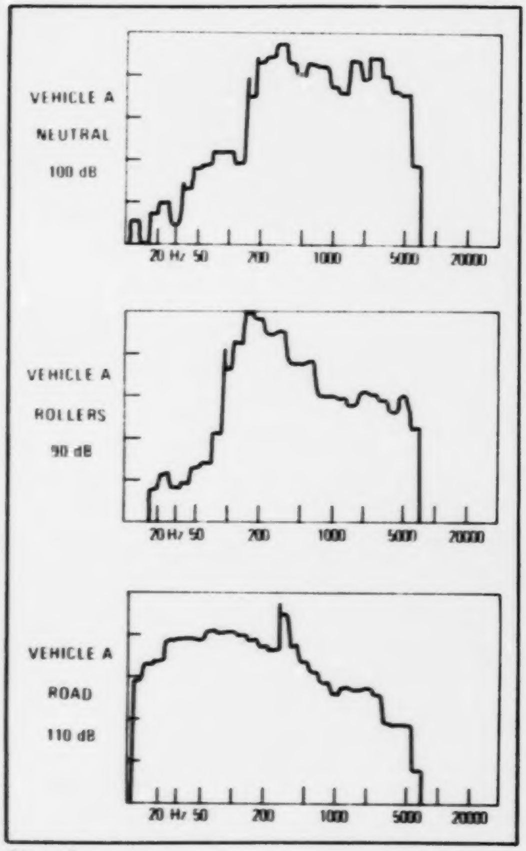

Figure 6

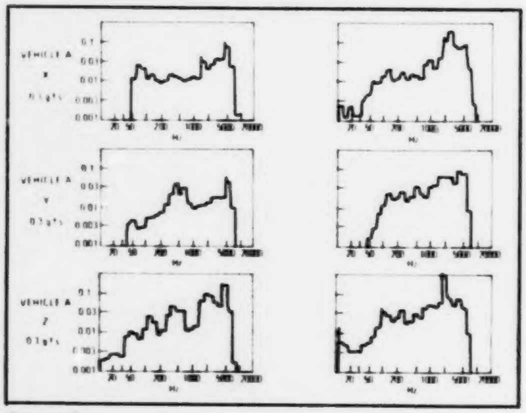

Figure 7

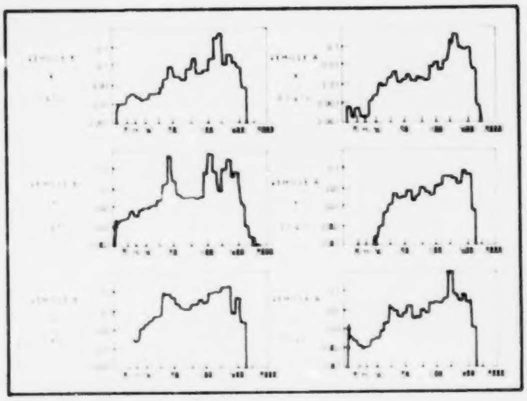

Figure 8

dangerous levels has been ascertained, it seems appropriate to continue with a fuller and more exhaustive investigation, which will also consider other working conditions of the vehicle.

CONCLUSIONS

The investigation carried out was intended to be only a first approach to the problem under consideration. In other words, it has enabled us to obtain some first stark facts about the low frequencies found in a motor vehicle, their respective amplitude and their distribution in the various situations analysed. It is intended to be the first part of a full statistical investigation which may be extended to all other fields of frequency up to the ultrasonic.

For this purpose, ISAM is working on a vibrating platform to simulate the various frequencies and intensities of vibration, in order to complete the examination of the problem from the physiological point of view.

THE ITALIAN TECHNICAL PRESENTATION

SECTION TWO

PART 7

FURTHER RESEARCH ON THE DRIVER/VEHICLE SYSTEM. STUDY OF A ROUGH–SURFACE STEERING PAD

LUCIANO CHIDINI
Presented by
ANGELO SCHIEPPATI, *Senior Active Safety Engineer*
Alfa Romeo S.P.A.

INTRODUCTION

In earlier international conferences on the safety vehicle we demonstrated the basic criteria which inspired Alfa-Romeo's activity in the field of Active-Safety. In particular we described our research project on the driver/vehicle system, and a study of the definition of a surface having a "rough" profile for road-holding tests. We will now describe the more significant progress attained in this research.

FURTHER RESEARCH ON THE DRIVER/VEHICLE SYSTEM

During the Third International Conference at Washington, we showed the results of our work in this field, in which a mathematical model of the driver/vehicle system was studied during a change of lane maneuver on a straight road; five different vehicle configurations being used at a speed of 70 mph (112.63 Km/h). The calculated results obtained were in agreement with the experimental ones. This congruity was also found in other cases of simple maneuvers, in the course of which the transverse acceleration of the vehicle has always been of modest proportions.

Successive experiments, however, involving maneuvers of particularly high intensity, both from the point of view of vehicle dynamic stress (maneuvers at high speed and acceleration, and extensions to the study of sports vehicles) and from the point of view of practical difficulties (sudden overtaking, "S" bends, etc.), have shown some non-acceptable discrepancies between the results obtained with the mathematical model, and those obtained experimentally.

Additional studies have proved that the cause of these discrepancies was to be looked for not only in some hitherto acceptable simplifications for the schematic representation of the vehicle, but also in the inability of the mathematical model of the driver to conform exactly to a real-life driver, at least in the above mentioned hard driving conditions.

The first cause of these errors is due to the imperfect knowledge of the dynamic behavior of the tire when operating in conditions close to its adhesion limits; added to which is the inevitably erratic behavior under such conditions.

Other causes of error are due to unforeseen structural elasticity of the vehicle and steering system, often considerably affecting both the dynamic load transfer and the steering characteristics of the front and rear suspensions. We do not intend to dwell on such refinements to the conception of the mathematical model of the vehicle, which do not present difficulties even if requiring a considerable amount of theoretical and experimental research. However, the description of that part of the mathematical model representing the driver's behavior is far more interesting and is still undergoing development.

Basically, the more recently introduced modifications allow the impressions the driver obtains from the road to be more realistically considered, without altering the model's essential characteristics of simplicity.

DESCRIPTION OF THE MATHEMATICAL MODEL OF THE DRIVER/VEHICLE SYSTEM AT ITS PRESENT STAGE OF DEVELOPMENT

It is generally assumed that the driver intends to keep his vehicle at a safe distance both from the curbs and from other reference points. Of course, such points may be either fictitious (i.e. imagined by the driver) or else real (e.g. lane markers.) A different intensity of steering action will therefore follow according to the type of driver. For this reason then, while trying to obtain maximum performance, the above mentioned reference points are reduced to a trajectory optimized in a way compatible to the limits imposed by the road edges; in other cases the driver acts on the steering only when the vehicle tends to pass the reference points on one side or the other. It is therefore assumed that the driver will generally estimate the following:

a) The deviation of the actual trajectory with respect

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to both the reference points assumed and the curbs.

b) Available transverse adhesion.

Moreover, the driver is characterized by two parameters:

c) Errors in evaluating the deviation of the actual trajectory with respect to both the reference points assumed and the curbs.

d) Reaction time.

The vehicle time constant and the driver's reaction time, make him adjust the vehicle's trajectory not on the instantaneous position, PI, but on a future position, PF, extrapolated in a time interval ΔT. While trying to avoid passing either the assumed reference points on the road, and what is more important, the curbs, the driver continuously carries out steering corrections until, as far as possible, the vehicle travels within the assumed reference points.

The maximum steering angle assumed for a given maneuver is that at which the adhesion limits of the front suspension is reached. If the driver were capable of estimating exactly the deviation of the effective trajectory with regards to the various reference points, he would act on the steering wheel in steps which would naturally be modulated by his reaction time. In actual fact, the driver's estimates are affected by errors.

On the actual road, the extrapolated trajectory of the vehicle is that shown in Figure 1, where the reference points 1 are physically defined and not passable (e.g. guard rails); the reference points 2 are also real, but do not represent an absolute limitation in that they consist of lane markers; the reference points 3 represent the corridor within which the driver intends to keep the vehicle trajectory, and hence are not physically defined.

Starting with the general situation described in Figure 1, the following cases, of practical interest, may occur:

a) The absence of reference points 2 (e.g. lack of lane markers), in which case the only physically defined reference points are the curbs; mathematically this is expressed by equating reference points 2 and 1.

b) The corridor, consisting of reference points 3, coincides with reference points 2; this situation occurs when the driver intends to use all the lane width.

c) The width of the corridor defined by reference points 3 can be neglected; this situation occurs when the driver intends to keep a trajectory which is as defined as possible. If to this condition we add that occuring when reference points 2 are widened to make them coincide with points 1, then we have the classical condition of driving on a racing circuit (trajectory to obtain maximum performance).

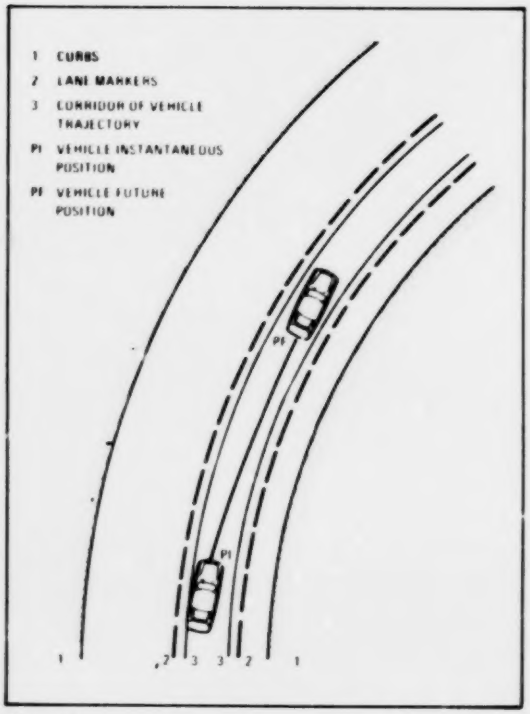
1 CURBS
2 LANE MARKERS
3 CORRIDOR OF VEHICLE TRAJECTORY
PI VEHICLE INSTANTANEOUS POSITION
PF VEHICLE FUTURE POSITION
PF
PI
1
2
3
3
2
1

Figure 1

In any case, the driver constructs reference points 3, basing them on the nearest physically defined ones (i.e. reference points 2). Naturally, in the absence of road markers as we have already mentioned, the only available reference points are the curbs.

The driver carries out a large number of estimations on the position of the future point within time intervals comparable to the system's time constants. Therefore, it is possible to introduce, as the resultant driver's action, the average of the effects of such estimates, distributed as a Gaussian function. The steering action of the driver is then assumed to be the integral of the product of the steering step function multiplied by a Gaussian function whose maximum value is centered on the correct value of the future point.

The steering step function (Figure 2) includes an interval of insensitivity in correspondence with reference points 3, which delimit the corridor within which the driver does not intend to perform steering corrections. As can be seen, the steering step inputs are different because of the variation in the steering angle necessary to reach the adhesion limits of the front suspension in either direction.

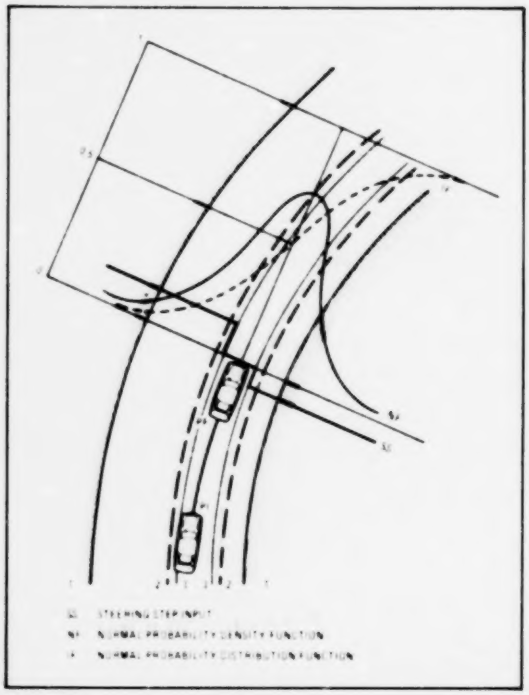
0.5
0
NF
SS
PF
1
2
3
3
2
1
SS STEERING STEP INPUT
NF NORMAL PROBABILITY DENSITY FUNCTION
IF NORMAL PROBABILITY DISTRIBUTION FUNCTION

Figure 2

The form of the Gaussian function that defines the steering angle obviously depends on the ability of the driver: the smaller his errors the greater will be the concentration around the average value of the function.

The concepts mentioned above are not only applied to reference points 3 (Figure 1), but also to the curbs. It has then been assumed that the evaluation errors with unreal reference points (e.g., a future trajectory arc, etc.) would be about twice greater than those made with easily visible ones. To be more precise, at any given instant, the steering angle is given as the appropriately chosen combination of the following terms:

Contribution Of The Curbs

The most dangerous point on the future trajectory arc contributes with the product of the maximum possible steering angle (in the opposite direction to the curb) and a reduction coefficient depending on the distribution of the estimate frequency of the distance between the curb and the future trajectory (Figure 3).

The driver's estimate of the most dangerous point is such that it need not necessarily coincide with the point actually nearest the curb. Generally, his evaluation of the most dangerous point is that for which the probability of being nearest the curb is at its greatest; this probability, naturally, depends on the size of the estimative errors made by the driver.

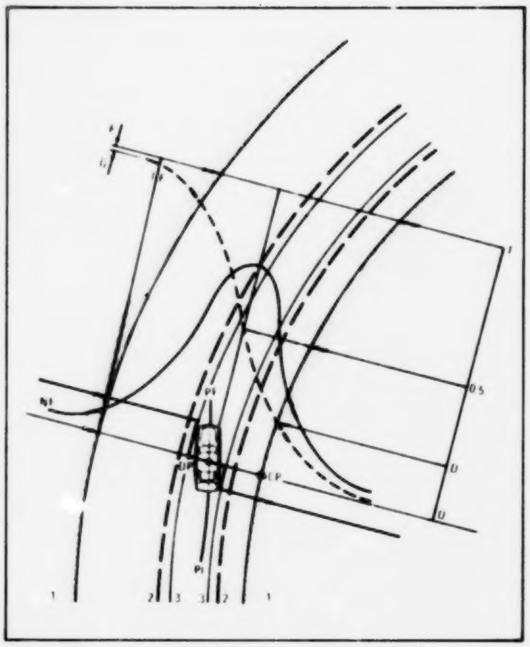

Figure 3

The reduction coefficient multiplying the maximum steering angle, as we have already shown, is the integral of the frequency distribution in the interval − ∞ CP. (Figure 3). This interval is represented by the segment OD, and gives the cumulative probability that the car, in the position DP, is in actual fact off the road. For symmetry, the contribution of the furthest curb is similarly evaluated giving a reduction coefficient indicated by the segment FG.

Contribution Of The Extrapolated Future Position

The driver's estimate of the future point position PF (Figure 4), with respect to reference points 3, may contain an error. The total contribution of the steering angle caused by the variation of this point is divided in two terms:

a) The contribution concerning the right hand reference 3, equal to the product of the maximum steering angle (in the sense deviating from the right hand reference), and a reduction coefficient depending on the frequency distribution of the estimates of the distance between the point PF and the right hand reference 3. This coefficient is the integral of the frequency distribution and gives the cumulative probability that the vehicle, in

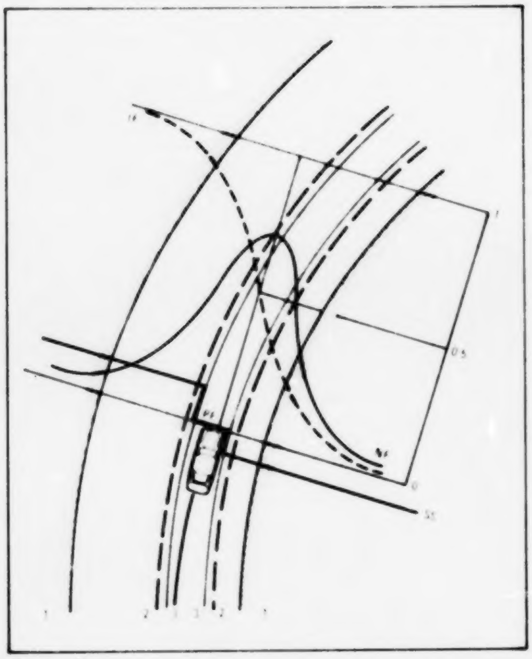

Figure 4

position PF, has actually passed the right hand reference point.

b) The contribution concerning the left hand reference 3 is similarly evaluated.

Generally, the contributions relative to the two references will differ, because of the asymmetry of the steering step, and because of the different frequency distribution of the estimate of the distances between PF and the references being examined.

To define the steering angle according to the above criteria, it is necessary to calculate both the curved segment of the trajectory extrapolated between PI and PF, and also the deviations σ in the driver's estimates.

The extrapolated curved segment PI-PF is calculated as an arc of a circle osculating the trajectory of the vehicle at the point PI, when the extrapolation distance has been defined. The deviations σ must also be calculated as a function of the extrapolation distance, of the curvature of the trajectory, and of the driver's characteristics. The parameters characterizing the driver influence the calculation of the deviation σ_i of the generical point on the future trajectory are according to the expression

$$\sigma_{(i=1,n+1)} = \frac{\left[(u \cdot \Delta T) \cdot \frac{i=1}{n}\right]^3}{R \cdot D} \qquad (1)$$

where:

u = The vehicle's velocity.
T = The extrapolation time interval. A function of the vehicle time constants and of the driver's reaction time.
R = The vehicle's trajectory radius of curvature.
D = The driver's characteristic distance, as specified beneath.

For references 3 we have assumed a deviation σ proportional to the distance between references 3 and 2, using experimentally obtained proportion coefficients.

Summing up, the contributions of the steering angle thus obtained can be evaluated according to the expression

$$\delta_t = a\,\delta_b + (1-a)\,\delta_f \qquad (2)$$

where:

δ_t = the resultant steering angle
δ_b = the steering angle due to the contribution of the curbs
δ_f = the steering angle due to the contribution of the future position
a = the cumulative probability of the vehicle leaving the road.

The Driver's Characteristic

The driver is easily characterized by two parameters:

a) The value of D affecting the equation for σ previously shown. It must be remembered that an increase of this distance corresponds to smaller values of σ or, in other words, improving the ability to evaluate the future point position PF with respect to the various references considered.

b) The reaction time. This affects the evaluation of σ only at a later stage, and especially in the definition of the real steering angle given by the driver, according to the expression:

$$T\delta + \delta = \delta_t \qquad (3)$$

where:

T = the driver's reaction time
δ = the real steering angle
δ_t = the resultant steering angle calculated by the expression (2)

EXPERIMENTAL VERIFICATION OF THE MATHEMATICAL MODEL OF THE VEHICLE/DRIVER SYSTEM.

We carried out a large number of tests involving two basic maneuvers (overtaking and "S" bend) at various speeds, three different types of Alfa Romeo vehicles, and drivers of different ability.

The only results taken into consideration were those obtained after the driver had become fully familiar with the test routine, the reason for this being that of improving test repeatability as much as possible, and anyway we found that full adaptability to both the maneuver and the vehicle itself was almost immediate, even for moderate ability drivers.

In order to save time, we will describe here the results obtained with only one type of maneuver (overtaking), one type of vehicle, and two drivers of different ability, driving at very similar speeds.

Description Of The Overtaking Maneuver

The driver carries out this maneuver at the predetermined constant velocity along the path described in Figure 5; the velocity is checked in various points of the path by means of photocells.

The test is performed in both directions, and the driver is allowed to use the full width of the lane, but without passing over the dashed lines limiting it (we preferred to avoid the use of lane marker cones, which would give the driver visual information of uncertain correspondence to any of the information the driver would actually get from the road. (See Model. . . . . page 417).

Vehicle Test Equipment

This consisted mainly of the following:

- A platform stabilized with a vertical axis gyro having 2 degrees of freedom. This platform allows us to measure the component accelerations along the longitudinal, transversal and vertical vehicle axes, and also the pitch and roll angles. By integration, it is then possible to obtain the longitudinal and transversal components of velocity and displacement.
- A rate gyro, to measure the angular velocity of the vehicle about the vertical axis.
- A radio receiver with an aerial at a low ground height; this apparatus relates the information obtained from the platform to the ground reference points, which are low frequency transmitter aerials just under the ground surface.
- A recording unit, either on magnetic tapes or on paper, according to the need.

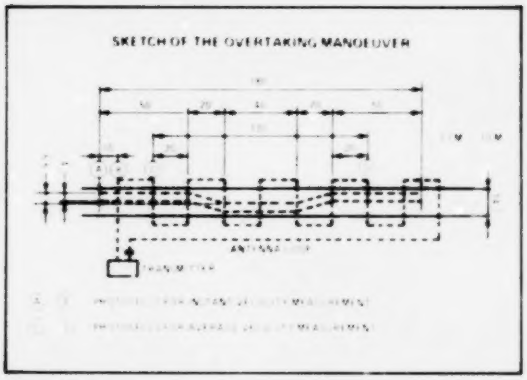

Figure 5

Characteristics Of The Vehicle, And Its Schematization

For this work, the vehicle is considered along the following scheme, which is the simplest possible in these cases; as the test is a constant velocity one, we have represented the vehicle with the following 4 degrees of freedom:

- The first degree of freedom is related to the law of motion of the center of gravity, filtered of the component of the roll motion. We assume the component of the velocity (relative to the longitudinal axis of the vehicle) to be constant.
- The second degree of freedom refers to the vehicle rotation about a vertical axis.
- The third degree of freedom is related to the roll motion.
- The fourth degree of freedom refers to the steering angle of the front wheels.

The tire reactions are considered non-linear with respect to both the slip angles and also the vertical loads.

The following are the data characterizing the tested vehicle:

total mass of the vehicle	139.5 $Kgsec^2/m$
sprung mass	122.16$Kgsec^2/m$
wheelbase	2.57 m
distance of the vehicle center of gravity from the front axle	1.305 m
distance of the vehicle center of gravity from the rear axle	1.265 m
front track	1.32 m
rear track	1.27 m
height of the center of gravity of the vehicle	0.5 m
distance of the center of gravity of the sprung mass from the roll axis	0.33 m
front roll center height	0.02 m
rear roll center height	0.38 m
front unsprung mass	7.14 $Kgsec^2/m$

rear unsprung mass	10.20 Kgsec²/m
height of the center of gravity of the front unsprung mass	0.286 m
height of the center of gravity of the rear unsprung mass	0.286 m
vehicle moment of inertia about the yaw axis	220.4 Kgmsec²
moment of inertia of the sprung mass about the roll axis	53.75 Kgmsec²
product of inertia of the sprung mass for yaw and roll axes	29.2 Kgmsec²
front roll stiffness	2780. Kgm/rad
rear roll stiffness	732. Kgm/rad
front roll damping	202. Kgmsec/rad
rear roll damping	92. Kgmsec/rad
static camber on front and rear wheels	0.
front wheels camber variation per unit roll angle	0.92
rear wheels camber variation per unit roll angle	0.
front roll steering	0.11
rear roll steering	0.

Equation for the side force F for the tire

$$F = \frac{3}{2}\mu\,\frac{\alpha}{\alpha_m}\left[1 - \frac{1}{3}\left(\frac{\alpha}{\alpha_m}\right)^2\right] P$$

where

μ = limiting coefficient of friction = $\mu_o\left(1 - \frac{P}{2\,P_m}\right)$

μ_o = coefficient of friction for $P = 0$ or $P_m = \infty$

α = slip angle.

α_m = slip angle which, for P = constant, gives the max. lateral force.

P = vertical load on the tire.

P_m = vertical load which, for α = constant, gives the max. lateral force.

for the front tires: $\mu = 0.9$ (for static load)
$\alpha_m = 13°$
$P_m = 500$ Kg

for the rear tires: $\mu = 0.9$ (for static load)
$\alpha_m = 10°$
$P_m = 500$ Kg

front axle lateral force variation per unit camber angle	673. Kg/rad
rear axle lateral force variation per unit camber angle	695. Kg/rad

Equation for the rolling resistance R of the vehicle:

$$R = P\,(a + b\,u^2)\quad (Kg)$$

P = vehicle weight (Kg)

a = 0.01075

b = $6.5 \cdot 10^{6}$ sec²/m²

u = vehicle forward speed (m/sec)

max. vehicle cross sectional area	1.7 m2
aerodynamic resistance coefficient	0.414
front aerodynamic lift coefficient	0.218
rear aerodynamic lift coefficient	0.019

Check of the correspondence between the results obtained with the vehicle mathematical model and those given by experimental tests. The data shown above were obtained from laboratory and test track experiments, these latter consisting of steady state steering input, etc.). As verification of the attendability of the above data, we show in Figures 6 and 7 a comparison between the calculated and experimental response to a given steering angle law related to an overtaking maneuver at 90 km/h. This comparison is carried out with the transversal accelerations and angular velocities. As we can see, the correspondence between the experimental and calculated values, even with a very simplified vehicle schematization, is fairly satisfying, notwithstanding the complication of the experimental law for the steering angle.

Check of the correspondence between the results obtained with the mathematical model of the vehicle/driver system and those given by experimental

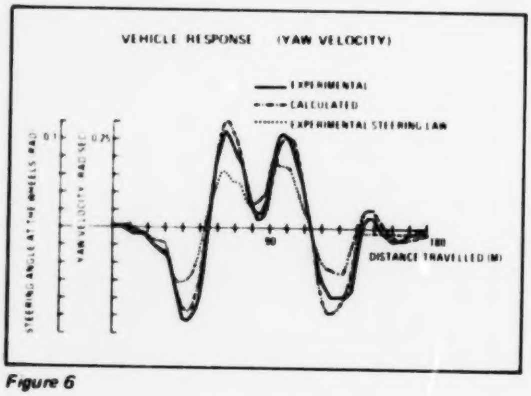

Figure 6

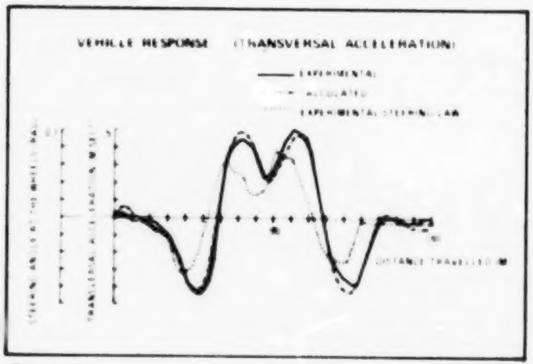

Figure 7

tests. For brevity, we will consider here the example given by two tests performed by two different drivers at comparable speeds:

Test 1 Driver A

Run	Speed Km/h
1°	87
2°	91
3°	88
4°	92

Average speed 89.5 Km/h

Test 2 Driver B

Run	Speed Km/h
1°	95
2°	95
3°	92
4°	92

Average speed 93.5 km/h

In Figures 8 and 9 we show the laws of the steering angles at the wheels obtained during the two tests. The analysis of this experimental data, carried out using a polynomial regression gave the following results:

Test 1 Driver A
Probability 90%
8th degree interpolation polynomial
Standard deviation 0.826°
Correlation coefficient 0.779

Test 2 Driver B
Probability 90%
7th degree interpolation polynomial
Standard deviation 0.975°
Correlation coefficient 0.638

The considerable scatter of the results, which characterizes these tests, hinders the close fitting of the regression polynomial to the average curve of the experimental phenomenum; we have high values of σ, and low correlation coefficients. The greater repeatability found in Test 1 (performed by a professional driver) is immediately noticeable with respect to that of Test 2 (carried out by an average, non-professional driver). Due to the above considerations, we considered it convenient to continue our work basing ourselves only on Test 1, and to compare the

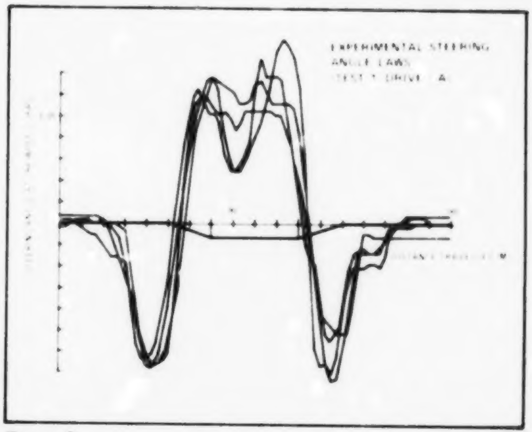

Figure 8

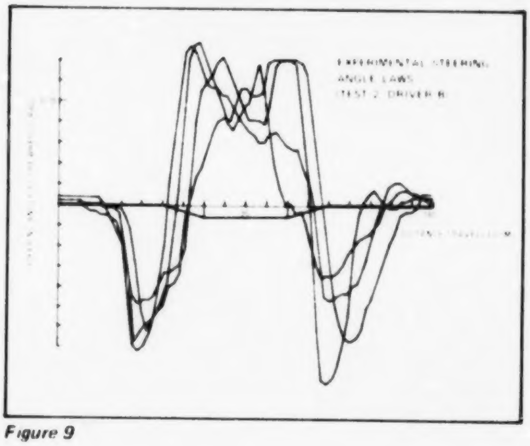

Figure 9

calculated results with the average curve of the experimentally obtained steering angles.

Referring, therefore, to Test 1, we tried to obtain the optimum value of the future point extrapolation distance (or time), as the estimated values concerning the behavior of driver "A" were known from a great number of previous experiments. Then

Reaction time 0.9 sec
Characteristic distance 400 m

The chosen value of the extrapolation distance (or time) is that giving the smallest mean square difference between the calculated steering law and the average experimental one. The optimum extrapolation distance found was 35.5 m, with a corresponding time of 1.42 sec.

Figure 10 shows the calculated law of the steering angles at the wheels superimposed to the average experimental one. The comparison is limited on the right hand side of the diagram to the last position of the vehicle for which the future point is within the dashed lines. The correspondence is fairly good, especially considering the scatter of the experimental results.

As the results obtained in the schematic representation of the vehicle were quite good (please refer to the Section on "Check of the correspondence between the results . . . . , page 422.) We consider it unimportant to relate here the comparison between the calculated and experimental values of the angular velocities and transversal accelerations.

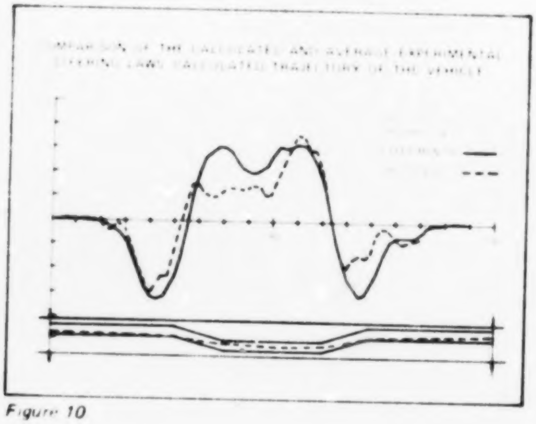

Figure 10

In Figure 10 we also show the calculated trajectory of the vehicle's center of gravity; the difference with the actual trajectory is small, and of the order of magnitude of the measurement errors.

Conclusions.

The mathematical model of the vehicle/driver system, in its present form, gives results which are fairly consistent with the experimental ones obtained from a large number of tests (of which, for brevity, we showed here only two examples). This correlation must be considered satisfactory if we keep in mind the substantial scatter of the experimental results, typical of such tests.

At present, the search for the extrapolation time is carried out by trial and error, minimizing the difference between the experimental and theoretical results. We have almost completed a research program which will allow us to relate the extrapolation time with the vehicle time constants. The results, till now, are very encouraging, but more work has still to be done on this.

The equation (page 420), expressing the law of the steering angles at the wheels, does not consider sufficiently the difficulty encountered by the driver in rapidly carrying out a maneuver requiring a large steering wheel rotation. We are now working on a modification to that equation which will also allow us to consider the inertial phenomena related to such a maneuver.

Finally, we are also researching on the generalization of the driver vehicle system mathematical model, to include the possibility of braking and accelerating inputs given by the driver.

To sum up, the simulation described above, in its present state, has already given useful results. Of course, as the phenomenon is very complex, the mathematical model will undergo further modifications, and hence become more complicated; we nevertheless believe that the basic structure of the mathematical model will remain unaltered.

SECTION TWO PART 7

PROGRESS IN THE DEFINITION OF AN UNEVEN SURFACE STEERING PAD FOR ROAD-HOLDING TESTS

LORENZO ROSTI ROSSINI
Presented by
ANGELO SCHIEPPATI, *Senior Active Safety Engineer*
Alfa Romeo S.P.A.

INTRODUCTION

Alfa Romeo showed, at the Third International ESV Conference, a first approach towards the definition of an uneven surface steering pad for standardized road-holding tests. We gave priority to steady state tests over transient ones, to emphasize the effect on road-holding of surface roughness only.

As far as the degree of roughness is concerned, we decided to reproduce the effect of a moderately uneven road. We therefore considered only the types of roughnesses described in Figure 1 and characterized by Power Spectra A, B and C shown in Figure 2.

Between the solutions shown in Figure 11, B and C were considered the most suitable. Solution C has the advantage of exciting a relatively wide range of frequencies, and of requiring the laying of a limited number of asperities onto the test surface.

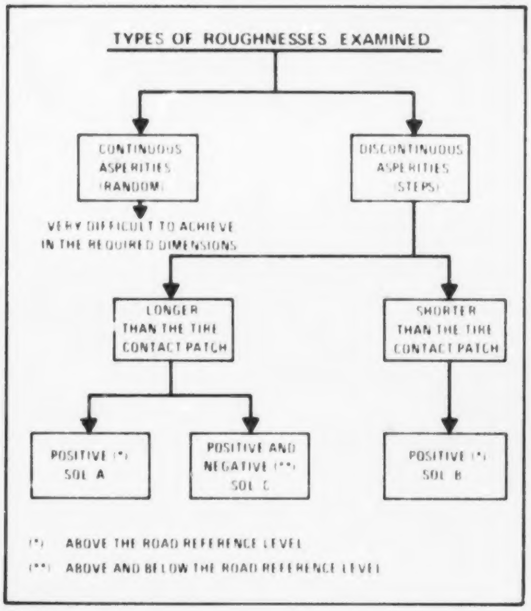
TYPES OF ROUGHNESSES EXAMINED
CONTINUOUS ASPERITIES (RANDOM)
DISCONTINUOUS ASPERITIES (STEPS)
VERY DIFFICULT TO ACHIEVE IN THE REQUIRED DIMENSIONS
LONGER THAN THE TIRE CONTACT PATCH
SHORTER THAN THE TIRE CONTACT PATCH
POSITIVE (*) SOL A
POSITIVE AND NEGATIVE (**) SOL C
POSITIVE (*) SOL B
(*) ABOVE THE ROAD REFERENCE LEVEL
(**) ABOVE AND BELOW THE ROAD REFERENCE LEVEL

Figure 1

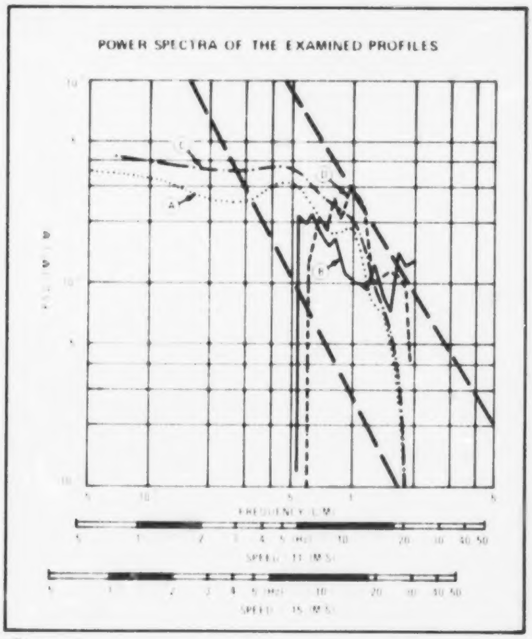

Figure 2

On the other hand, solution B allows to approximate closely to the road Power Spectrum in the range (6 + 18 Hz), most important for wheel/ground contact considerations. Moreover, such solution causes a more uniform excitation, and also allows to obtain an arbitrary out-of-phase of the various components of the excitation. For these reasons, then, solution B was reputed the most suitable for a complete and realistic examination of the average behavior of vehicles on the road. Therefore, work on experimental test surfaces based on such solution was started.

We intend to relate here the results of the tests carried out on such surfaces, and to suggest indications for a better definition of the possible characteristics of a standardized test surface.

TEST PROCEDURE

The tests were carried out on an asphalted, level, steering pad annulus, having 19 m internal radius, and 8 m. width. The roughnesses, represented by removable obstacles, were placed radially along two concentric annuli, each 2 m. wide, with the common circumference having 23 m. radius (Figure 3).

Each of the surfaces obtained with various types of obstacles were tested, in the clockwise and anticlockwise directions, by groups of vehicles consisting of between five to 12 cars, both of the same type and also of various types. The general characteristics, and some particulars of the suspensions (weights, springs, dampers and tires) were checked on the Alfa Romeo vehicles used. The cars were driven by different drivers, and the results were averaged over at least 10 laps, after ensuring that each driver had become fully familiar with the test.

At present, the results are the times taken to lap the annulus at the maximum possible speed. In addition to this data, analogical recordings of the vehicle behavior were obtained; we are still processing these results.

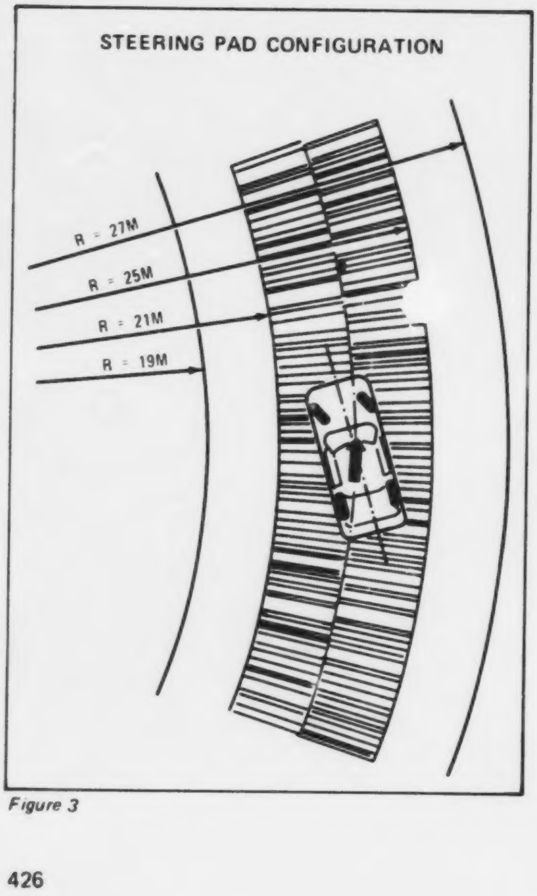

Figure 3

CHARACTERISTICS OF THE TEST ROUGHNESSES

From Figure 3 we can see that the test roughnesses consists of radially positioned bars, most of which have a rectangular cross section with rounded corners; the sectional area above the ground is between 2 and 4 cm^2 (e.g., in Washington we described the obstacles used for solution B).

Tests have also been carried out with equal obstacles; one such test was that shown to the NHTSA representatives during their visit to our Balocco test track on September 18, 1972.

As far as the material used for these obstacles is concerned, we have had a lot of problems with steel bars anchored to the asphalt surface. Indeed, apart from the different coefficients of friction of steel and asphalt, we found that the characteristics varied during the tests, due to the oxidation of the bars. We therefore believe it advisable that the obstacles and the rest of the test surface be made of the same material, for example concrete.

CHARACTERISTICS OF THE STEERING PAD

We consider the dimensions of the steering pad and the amplitudes of the obstacles we used quite adequate (particularly those of solution B), for the requirement of having a reasonably sized disturbance and also a fairly low vehicle test speed.

On the other hand, we have met with some difficulties on the steering pad having the two separated annuli (Figure 3):

- It is a hard task for the driver to follow the given circular trajectory separating the two annuli, whereas he prefers to move as near as possible to the inner circumference of the smaller annulus.
- Due to the lanes' curvature and the yaw angle of the vehicle, it is very difficult to avoid having one of the wheels travel on the circumference of separation of the two annuli. When this occurs, the wheel is at the same time excited by the inner and outer obstacles.
- In crossing over from one annulus to the other, the tire hits the ends of the bars, with the possibility of damage to the tire walls.

The above reasons imply the need for continuity of the obstacles from one end of the steering pad to the other, so that the possibility of exciting independently the left hand and right hand wheels is greatly limited.

REPEATABILITY

We have already mentioned the need for the driver to be perfectly familiar with the test and with the vehicle: but even when this is the case, there will be errors, which have to be added to those due to the variations in the friction coefficients of the surface (caused by the rubber film deposited on it) and of the obstacles.

If these problems are not taken into account then, over a day's testing, errors of up to 15% on the maximum accelerations of a given vehicle may result.

On the other hand, by avoiding as much as possible the errors due to driver training for this test, then a good repeatability of the measured accelerations will occur (with errors smaller than 2%, even when carrying out tests after some time interval.

In any case, we foresee the possibility of performing tests on a wetted steering pad, in order to also reduce the problem caused by the rubber film deposits; such improvement, added to the one afforded by making the surface and the obstacles of the same material, and to thorough driver training, should result in a very good test repeatability.

TEST SELECTIVITY

From this point of view, the tests have been satisfactory. For example, if we examine only the results obtained with Alfa Romeo vehicles (therefore with vehicles characterized by fairly homogeneous standards of road-holding and adhesion), we find (Figure 4) that the transversal accelerations obtained on the surface with steel bar obstacles (type D, Figure 2) are in the ratio 1.19 between the best and the worst vehicle of the same group, whereas on a smooth steering pad the equivalent ratio for the same group of vehicles is 1.12.

Generally, we found that the selectivity on a rough surface is equivalent to, or only slightly higher than, that on a smooth surface, at least for vehicles having well tuned suspensions. For conditions other than that, the selectivity on a rough surface is considerably greater. On one Alfa Romeo vehicle we also carried out a series of tests with dampers adjusted rather differently to those of production vehicles. The rating of the dampers tested are described with reference to production units in Figure 5. Correspondingly, in the same figure are shown the maximum accelerations obtained on the type D steering pad. It can be seen that, for relatively modest adjustment variation (exceeding only slightly the tolerance on production dampers), the maximum accelerations vary by less than 4%. This is due to the fact that the adjustment of production dampers is already optimized; considerably greater variations are obtained with dampers set very differently from the optimum. Analagous tests, on smooth ground, did not show any appreciable variation.

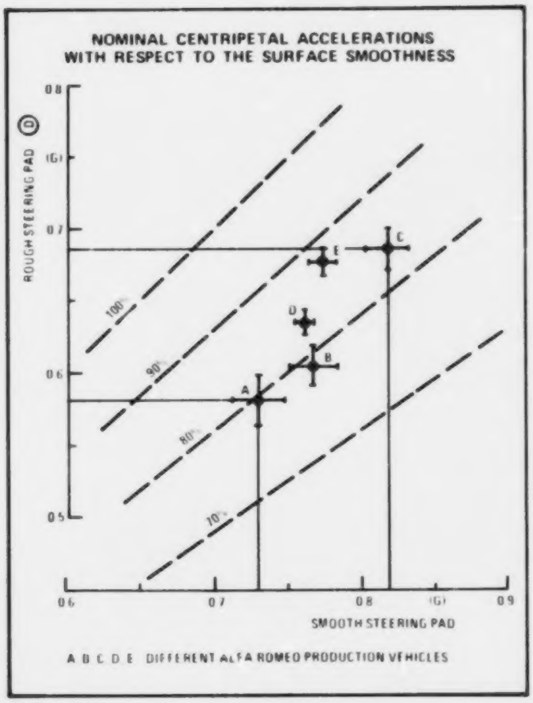

Figure 4

SCATTER OF THE RESULTS

As can be seen in Figure 4 (where, for each type of car tested, are indicated segments proportional to the mean square deviation of the acceleration obtained in ten runs), the scatter of the results taken on the rough surface is of the same order of amplitude as the one obtained on a smooth surface.

Moreover, testing a group of Alfa Romeo vehicles of various types on the type D surface, we found a correlation coefficient of 0.957 (with a 95% probability), between the scatter of the results obtained on the rough surface and that obtained on a smooth surface (Figure 6). This could prove that the scatter of the results is a parameter mainly dependent on the vehicle itself (or, more precisely, on the driver/vehicle system), and only to a lesser degree on the type of test surface. The order of magnitude of the mean square deviation of the accelerations obtained seems to us, on the other hand, rather high in comparison to the differences between the maximum accelerations of various types of vehicles; we would therefore tend to propose a larger tolerance when deciding on the

INFLUENCE OF THE REAR DAMPERS ON A PRODUCTION ALFA ROMEO VEHICLE

DAMPER SETTING	ACCELERATION (G)	ACCELERATION RATIOS	R.M.S. ACC (G)
PRODUCTION	0.660	1	0.010
A	0.670	1.015	0.010
B	0.646	0.979	0.012

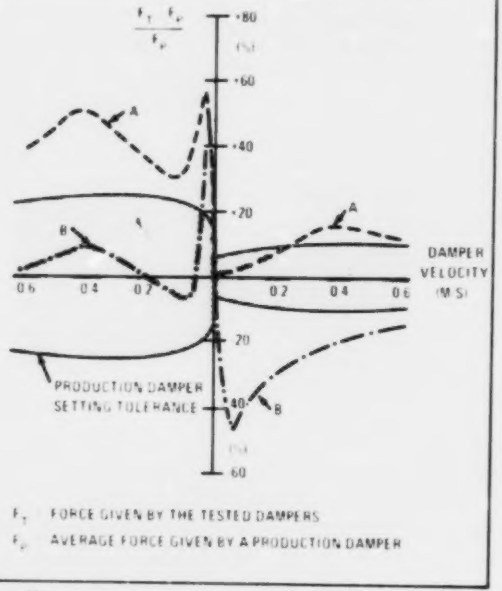

Figure 5

lower limit for maximum transversal acceleration of "safe" vehicles.

THE CORRELATION BETWEEN A SMOOTH AND ROUGH SURFACED STEERING PAD

Figure 4 gives a sufficiently significant example of what we have found with our tests.

An analysis of the correlation between the accelerations obtained on a rough and smooth steering pad was carried out. We found that the correlation test was satisfied, when assuming a linear correlation, only if the correlation probability was limited to 65%.

If we consider that this result has been obtained using a group of different Alfa Romeo vehicles, which were, however, similar in performance and tuning characteristics, we can then confirm that tests carried on rough steering pads and on smooth steering pads give, in general, independent results.

Besides, a reduction in the maximum transversal acceleration is always found when passing from a smooth to a rough surface; for a type D surface, this changes from a minimum of 10% to a maximum of 30% for vehicles having good adhesion on rough surfaces.

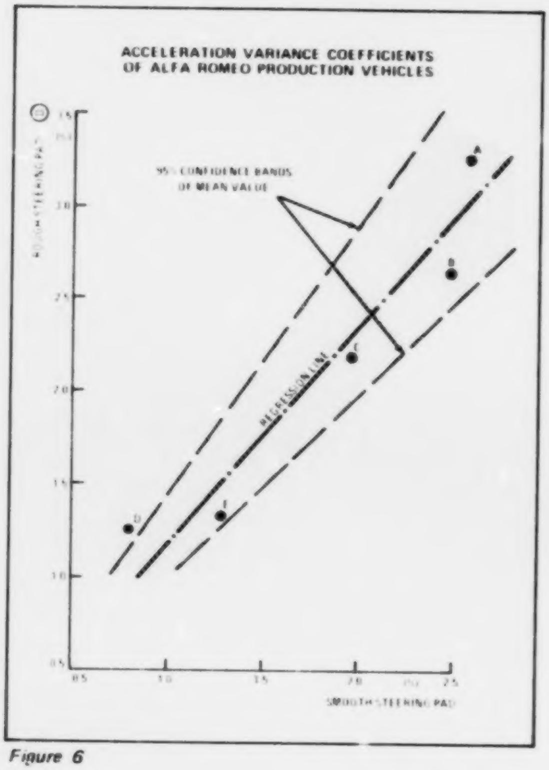

Figure 6

We consider this reduction to be sufficient; to increase it, the velocity and the obstacle cross section would have to be increased.

OTHER EXPERIMENTAL RESULTS

As we have already mentioned, in addition to time measurements over a fixed basis (and hence the calculation of a nominal transversal acceleration), we have carried out other measurements on the motion of the vehicles and on the steering angles. This has been done principally to study the oversteer characteristics and the vibrations of the vehicle body with respect to passenger comfort.

Under this aspect also, the various types of test surfaces have shown themselves to be sufficiently valid, and it has been confirmed that a vehicle may show completely different characteristics both over and understeer) on a rough and on a smooth steering pad.

Even for this kind of tests the section of type B and D obstacles can be considered acceptable, at the velocity indicated in Figure 2.

DEVELOPMENT OF THE ROUGH STEERING PAD

In relation to the studies and tests of which we have supplied the principal results, Alfa Romeo is developing a new type of rough surface for steering pad tests. This surface consists of a single annulus (no longer divided in lanes) having the internal radius about 20 meters, and a width of 6 meters. The ring has "built-in" obstacles, is made of reinforced concrete, and has an inward gradient of less than 2% (for drainage purposes).

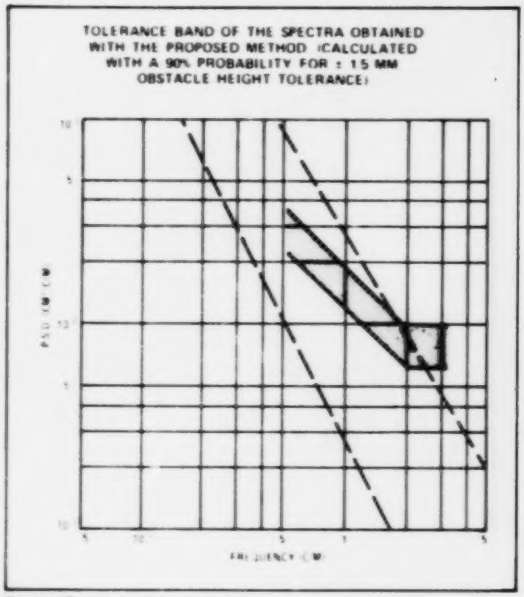

Figure 7

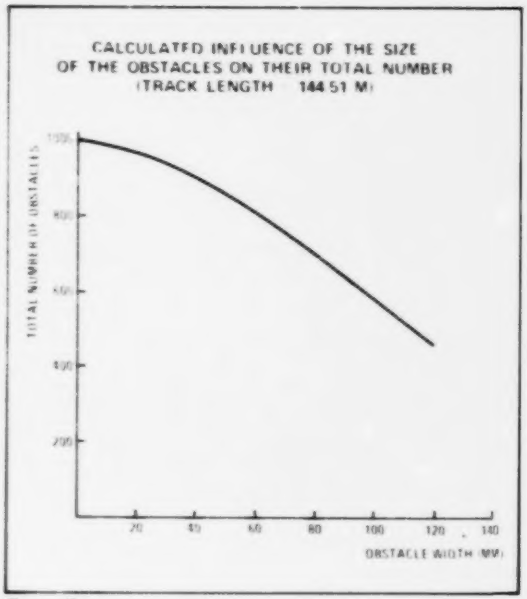

Figure 8

ROUGH SURFACE EXAMPLE
OBSTACLE HEIGHT Y(MM) AS A FUNCTION
OF DISTANCE TRAVELLED X (M)

X	Y	X	Y	X	Y	X	Y	X	Y
0.05	9	2.20	0	4.35	0	6.50	17	8.65	6
0.10	0	2.25	0	4.40	0	6.55	0	8,70	0
0.15	0	2.30	0	4.45	0	6.60	0	8.75	6
0.20	0	2.35	3	4.50	0	6.65	3	8.80	0
0.25	6	2.40	6	4.55	0	6.70	6	8.85	0
0.30	0	2.45	0	4.60	6	6.75	0	8.90	0
0.35	0	2.50	6	4.65	6	6.80	0	8.95	12
0.40	6	2.55	0	4.70	0	6.85	0	9.00	6
0.45	0	2.60	6	4.75	0	6.90	0	9.05	0
0.50	6	2.65	0	4.80	9	6.95	0	9.10	0
0.55	6	2.70	0	4.85	0	7.00	0	9.15	6
0.60	3	2.75	0	4.90	3	7.05	0	9.20	0
0.65	0	2.80	0	4.95	0	7.10	6	9.25	3
0.70	0	2.85	6	5.00	0	7.15	0	9.30	0
0.75	0	2.90	0	5.05	0	7.20	0	9.35	0
0.80	6	2.95	0	5.10	0	7.25	0	9.40	0
0.85	0	3.00	0	5.15	0	7.30	0	9.45	0
0.90	0	3.05	6	5.20	12	7.35	0	9.50	0
0.95	0	3.10	0	5.25	0	7.40	0	9.55	9
1.00	0	3.15	6	6.30	12	7.45	0	9.60	0
1.05	0	3.20	12	5.35	0	7.50	12	9.65	0
1.10	0	3.25	0	5.40	0	7.55	6	9.70	0
1.15	0	3.30	0	5.45	0	7.60	0	9.75	12
1.20	6	3.35	6	5.50	0	7.65	0	9.80	0
1.25	0	3.40	0	5.55	0	7.70	12	9.85	0
1.30	0	3.45	0	5.60	0	7.75	0	9.90	6
1.35	0	3.50	0	5.65	6	7.80	0	9.95	0
1.40	0	3.55	0	5.70	0	7.85	6	10.00	0
1.45	9	3.60	0	5.75	0	7.90	0	10.05	0
1.50	0	3.65	0	5.80	3	7.95	9	10.10	3
1.55	6	3.70	0	5.85	0	8.00	0	10.15	6
1.60	15	3.75	0	5.90	0	8.05	6	10.20	0
1.65	0	3.80	0	5.95	0	8.10	0	10.25	0
1.70	6	3.85	6	6.00	0	8.15	0	10.30	0
1.75	0	3.90	0	6.05	6	8.20	0	10.35	0
1.80	0	3.95	0	6.10	0	8.25	0	10.40	6
1.85	0	4.00	0	6.15	6	8.30	0	10.45	0
1.90	0	4.05	9	6.20	0	8.35	3	10.50	6
1.95	6	4.10	0	6.25	0	8.40	0	10.55	0
2.00	6	4.15	6	6.30	0	9.45	0	10.60	0
2.05	0	4.20	0	6.35	9	8.50	0	10.65	0
2.10	0	4.25	0	6.40	6	8.55	0	10.70	0
2.15	0	4.30	6	6.45	0	8.60	0	10.75	0

Figure 9

The obstacles consist of radial concrete slabs buried by a sufficient quantity, and protruding from the ground by amounts which vary in relation to the position of the obstacles.

The slabs have 3 mm. steps, to provide a sufficient fitting tolerance: it has indeed been found that, with such a tolerance, the spectrum of the road profiles is within the band indicated in Figure 7 with a 90% probability.

To such spectrum variation should correspond a variation in the maximum accelerations obtainable (for the same vehicle) within a 2% margin. The definition of the height of every roughness is based on a calculation which estimates the total impulse (in terms of the area of the projecting section) relative to each 50 mm. track length.

The total impulse is, in turn, estimated by superimposing individual periodic impulses, according to the criteria by us described in Washington last year. The area of the section obtained, divided by the 50 mm. base, gives the height of the generic obstacle; such height is afterwards rounded to the nearest multiple of the basic height of 3 mm. (if the height is zero, there is no obstacle).

The width of 50 mm. has been chosen in relation to the errors caused by the approximation, and the number of necessary obstacles. In fact, increasing this length, the number of necessary obstacles decreases (see Figure 8), but the spectrum error increases. In the following table we show a typical example of a profile defined by the height (in mm.) in relation to each following 50 mm. track length (Figure 9).

THE ITALIAN TECHNICAL PRESENTATION

SECTION TWO

PART 7

FRONTAL IMPACT – A THEORETICAL ANALYSIS OF THE HEAD AND NECK OF A PASSENGER WEARING SAFETY BELTS – THE STUDY OF THE EFFECTS OF A HEAD RESTRAINT SYSTEM

MARCO GARETTI
and
ITALO POLETTI
Presented by
ANGELO SCHIEPPATI, *Senior Active Safety Engineer*
Alfa Romeo S.P.A.

Impact Characteristics

This study concerns a 40 mph (64.36 km/h) frontal impact against a rigid barrier. The frontal deformation of the car measured with respect to the anchorage points of the occupant restraint system is assumed to be such that the maximum deceleration of the center of gravity of the torso will never exceed 60 g. The restraint system is also assumed to prevent head and torso impact against the interior of the vehicle, notwithstanding the survival space reduction caused by the crash.

To estimate the survival space, a rearward dashboard displacement of about 30 mm is assumed; this is brought to 100 mm to allow for the presence of the collapsed steering wheel, and the same value is also assumed for the rearward displacement of the foot well, near the pedals.

We will refer later to this deformed configuration of the vehicle, as shown in Figure 1.

A passenger compartment deceleration law filtered of all harmonic components higher than the first was also assumed, in accordance with the work shown at the Third International ESV Conference "Influence of the Seat Belts Anchorage Points Deceleration Mode on Severity Indices."

We would also like to add that the analysis of a considerable number of experimental deceleration diagrams of passenger compartments has brought us to the conclusion that, in most cases, the amplitude of the first harmonic of the Fourier series development is roughly equal to the constant component of the said development, so that the deceleration law can be written as follows:

$$x = -\frac{D}{2}\left(1 - \cos 2\pi\frac{t}{T}\right) \qquad (1)$$

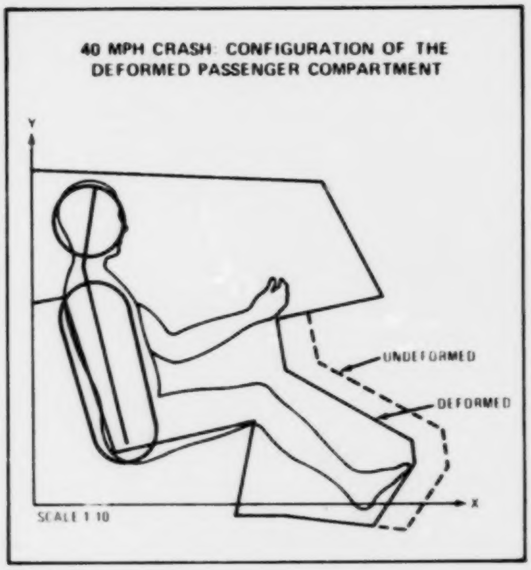
40 MPH CRASH: CONFIGURATION OF THE DEFORMED PASSENGER COMPARTMENT

Figure 1

where

x = passenger compartment acceleration (at the restraint system anchorage points)
D = deceleration peak
t = time measured from the start of the impact
T = impact duration

Integrating the above expression, the velocity and deceleration distance can be obtained:

$$\dot{x} = u\left[1 - \left(\tau - \frac{1}{2\pi}\sin 2\pi\tau\right)\right] \qquad (2)$$

$$X = L\left[\tau(2 - \tau) + \frac{1}{2\pi^2}(1 - \cos 2\pi\tau)\right] \qquad (3)$$

where

$\dot{x}$ = velocity at time t
u = initial impact velocity
x = displacement after time t
L = total deformation of the front of the vehicle
τ = t/T

$\dot{x}$, x and L always refer to the restraint system anchorage points. D, u, L and t have the following relationships:

$$D = \frac{2u}{T} \tag{4}$$

$$L = \frac{uT}{2} \tag{5}$$

therefore

$$DL = u^2 \tag{6}$$

Figure 2 shows non-dimensional curves of the decelerations, velocities and displacements calculated.

Equations (1), (2) and (3) describe, in sufficiently approximated form, the law of motion of the restraints anchorage points during the impact. Equations (4), (5) and (6) express the relationship existing between the principal parameters defining the impact conditions and allow a quick evaluation of the effects of impact speed and vehicle deformation on the deceleration peak.

For example, if we assume that $T \simeq$ constant as u changes, as can be verified experimentally, then we obtain that the frontal deformation and deceleration peaks are directly proportional to impact velocity. With respect to the data previously fixed for this work (i.e., impact velocity 40 mph, survival space as defined in Figure A1, and given seat belts configuration), it follows that we had to assume a vehicle frontal deformation of 550 mm, in order not to exceed, for the various cases examined, the limit of 60 g as the maximum deceleration of the torso center of gravity.

To sum up, the assumptions were:

u = 17.8 m/s
L = 0.55 m
and hence
T = 62 ms
D = 574 m/s^2

Description of the dummy

Given the type of impact, it will be assumed that the law of motion of the dummy can be described in the plane of Figure 1. Moreover, having to limit this research essentially to the study of head and neck stresses, we used a partial simulation of the dynamic behavior of the passenger, applying the 4 degree of freedom system shown in Figure 3. Comparisons between a high number of experimental and theoretical results has proved to us that this system, incorporating the lower and upper limb's equivalent masses, gives a sufficient description of the law of motion of the head and torso. The masses and dimensions are those of a 50 percentile male.

In order to carry out a reliable study of the head and neck stresses, it is necessary to know the dynamic

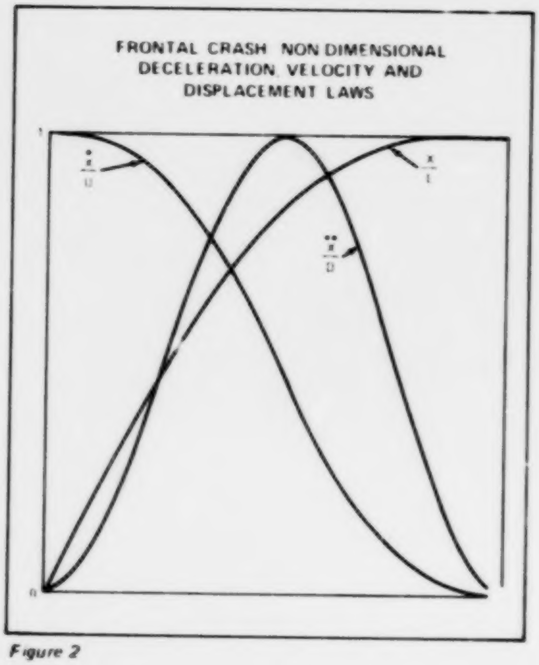

Figure 2

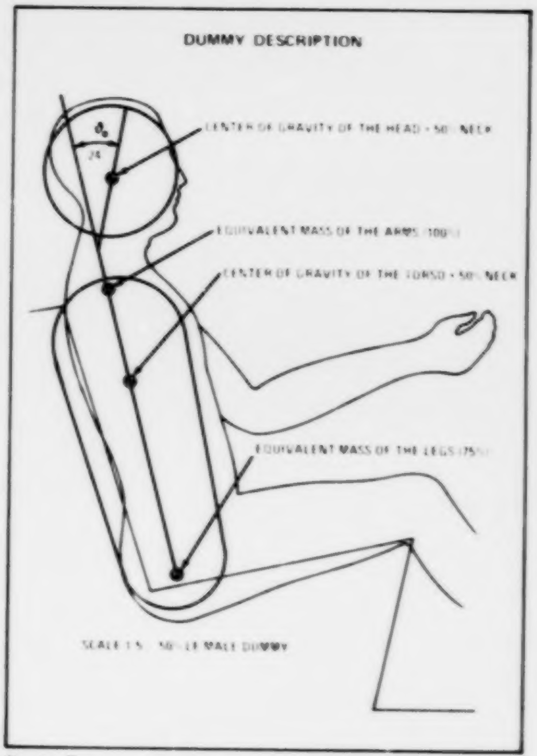
DUMMY DESCRIPTION
ϑ_0
24
CENTER OF GRAVITY OF THE HEAD + 50% NECK
EQUIVALENT MASS OF THE ARMS (100%)
CENTER OF GRAVITY OF THE TORSO + 50% NECK
EQUIVALENT MASS OF THE LEGS (75%)
SCALE 1:5 – 50%ILE MALE DUMMY

Figure 3

neck behavior in bending, both when the head is freely rotating, and also in the following phase, when rotation is hindered by the contact between the chin and the sternum (hyperflection).

It is particularly important to know the dynamic behavior of the system in this last phase, in which the greatest stresses are usually found. Unfortunately, very little research has been carried out in this field, occasionally with contradictory results. We have assumed that the law of the moment of resistance at the hinge simulating the neck is that shown in Figure 4.

LAW OF THE VARIATION OF THE MOMENT OF THE FORCES APPLIED BY THE TORSO TO THE HEAD ABOUT HINGE "O"

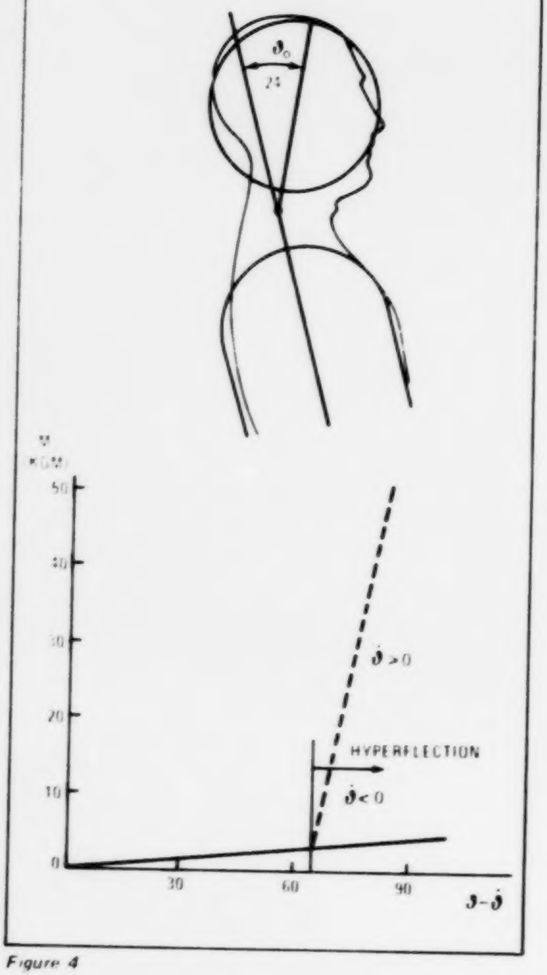

Figure 4

The full line indicates the reversible elastic contribution of the hinge, while the dotted line (in the hyperflection zone) shows the additional moment about the hinge, due to the reaction of the sternum on the chin.

We consider, therefore, such additional moment to be irreversible due to the trauma caused in the neck in the hyperflection phase, where we assume valid the law of moment shown as the full line for $\dot{\theta} < 0$, and the dotted line for $\dot{\theta} > 0$.

In both cases, the moment of the hinge itself is represented by the full line. Moreover, there is an additional damping moment at the hinge of the form

$$M = -\beta\dot{\theta}$$

where $\beta = 0.06$ Kgs m.

Due to the already mentioned uncertainties of the numerical data contained in Figure 4, the calculation results will be the more reliable the smaller the head rotation angle (with respect to the torso) in the hyperflection zone.

Safety Belts

The dummy is restrained by shoulder and lap belts as shown in Figure 5. The diameters of the contact circumferences of the lap and shoulder belts have been chosen as a function of the correct positioning of the belts relative to the body.

The dimensions of the torso remain, however, as shown in Figure 3.

The damped elastic elements giving a vertical and horizontal reaction on the point H represent the seat reaction.

All the belts are considered constant rigidity systems having no preload and time delay.

In accordance with our already mentioned paper read at the Washington Conference, we have assumed critically damped belts.

For each case considered, we have chosen a belt rigidity to exploit the available survival space, with a certain margin to avoid the danger of head and torso impact against the interior of the vehicle.

Device To Restrain Head Movement With Respect To The Vehicle Interior

The system consists of a curtain fixed inside the vehicle at an appropriate point behind the occupant's head. The curtain is positioned in front of the occupant's head so as to come into effect after about 40 ms from the beginning of impact.

To this time interval corresponds a forward enough positioning of the curtain to ensure that its deployment will take place with complete safety, even when the occupant is not seated correctly.

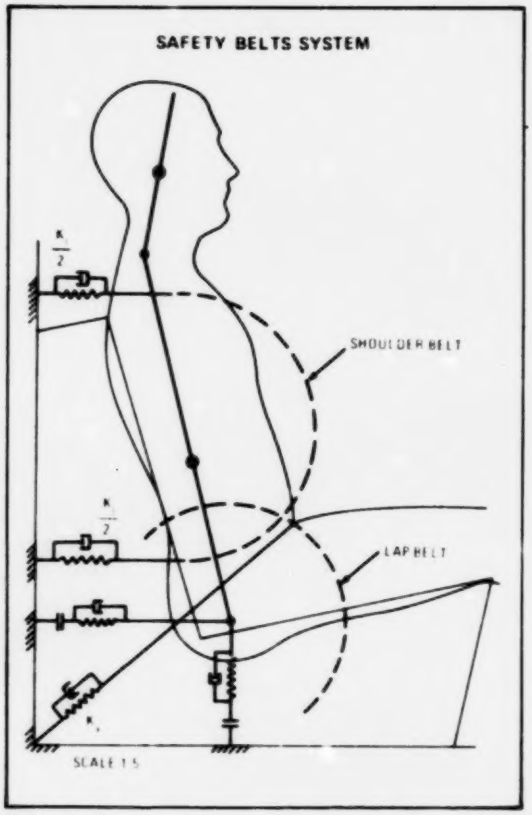
SAFETY BELTS SYSTEM

Figure 5

When the head contacts the curtain, there is a resulting head decelerating force passing through the curtain anchorage point.

The reaction is linearly elastic, and critically damped (Figure 6).

Analysis Of The Results Obtained

We have examined the following cases:

Case 1: Passenger with safety belts, without any head restraint device.

Case 2: Passenger with safety belts and with a head restraint device.

For each case we show the best results between all those obtained varying the rigidity of the restraint systems. In the following table we indicate the optimized rigidities, expressed in Kg/m.

	K_t	K_a	K
Case 1	7000	7000	0
Case 2	2000	6000	200

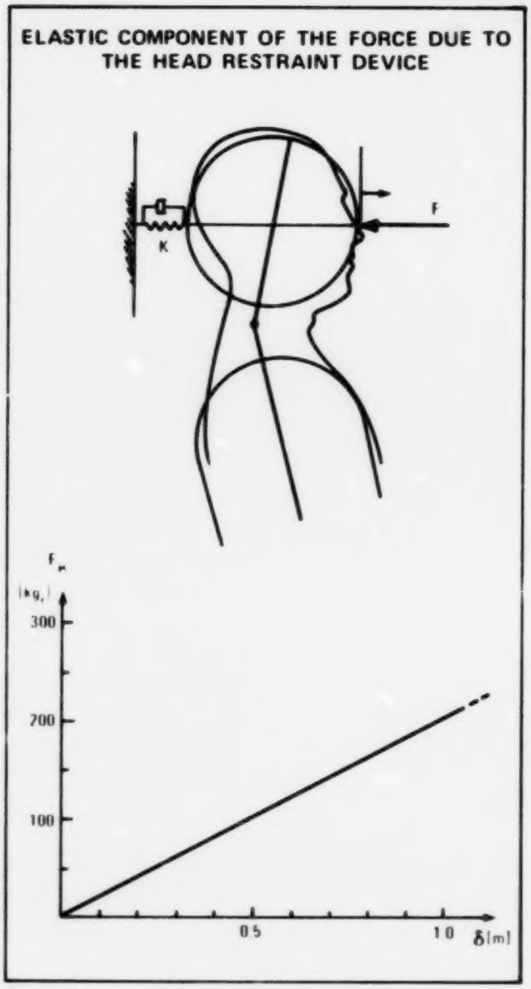
ELASTIC COMPONENT OF THE FORCE DUE TO THE HEAD RESTRAINT DEVICE

Figure 6

Figures 7 and 8 show the laws of motion of the head and torso in both cases, starting from the beginning of the crash. We can observe the considerable forward rotation of the head in Case 1 which, on top of giving a large hyperflection of the neck, also forces us to reduce the torso forward displacement, with respect to Case 2, by increasing the rigidity of the lap and shoulder belts, as can be seen from the above table.

In Case 2 there is no hyperflection of the neck.

In Figure 9 we compare the decelerations of the center of gravity of the torso in both cases. The deceleration peak is slightly greater for Case 1 than for Case 2, but it is lower than the given 60 g limit. In the next figures we compare, for both cases examined, the diagrams of the following quantities relative to the head motion and neck stresses.

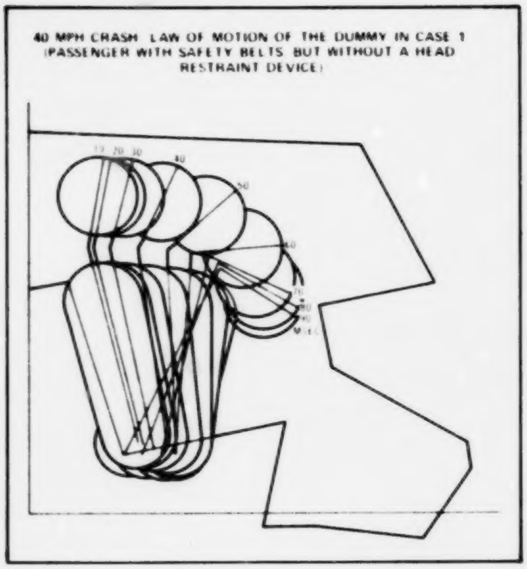
40 MPH CRASH LAW OF MOTION OF THE DUMMY IN CASE 1
(PASSENGER WITH SAFETY BELTS BUT WITHOUT A HEAD RESTRAINT DEVICE)
10 20 30
40
50
60
70
80
90
MSEC

Figure 7

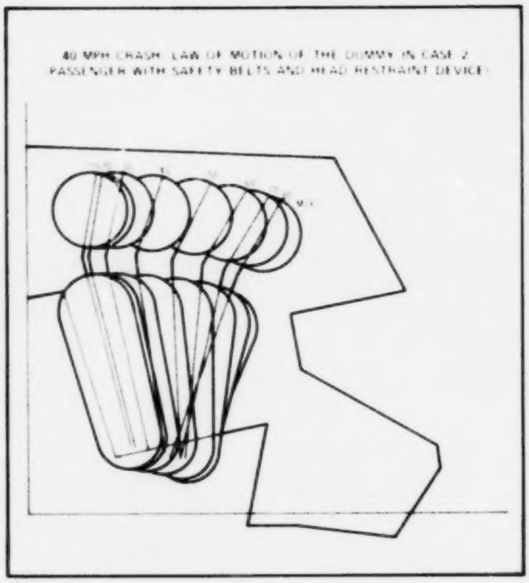
40 MPH CRASH LAW OF MOTION OF THE DUMMY IN CASE 2
(PASSENGER WITH SAFETY BELTS AND HEAD RESTRAINT DEVICE)

Figure 8

a_h Figure 10 Acceleration of the center of gravity of the head expressed in g.

ω Figure 11 Angular acceleration of the head, expressed in rad/sec^2 (clockwise positive).

SI Figure 12 Severity Index of the head evaluated with the formula

$$SI = \int_0^t a_n^{2.5}\, dt$$

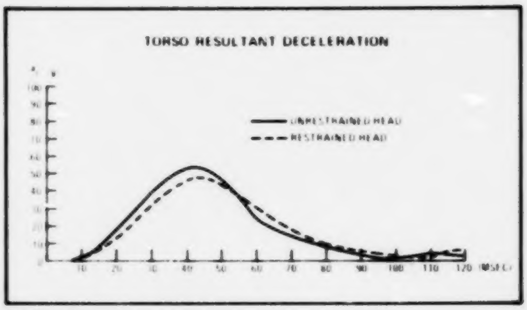

Figure 9

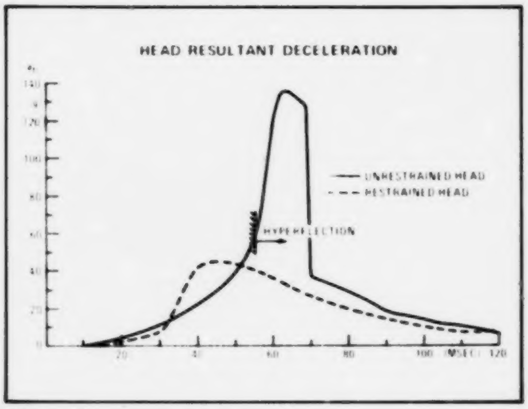

Figure 10

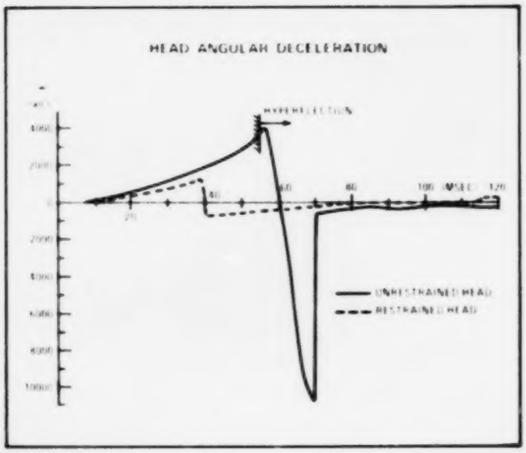

Figure 11

S Figure 14 Shear force (in Kg) applied to the hinge, and evaluated in the direction of the bisector of the angle between the head and torso axes (Figure 13); S < 0 when the head applies to the torso a force in the direction shown in the figure).

T Figure 15 Traction force (in Kg) applied to

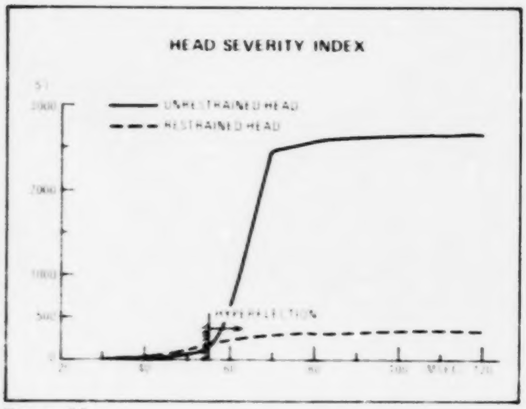

Figure 12

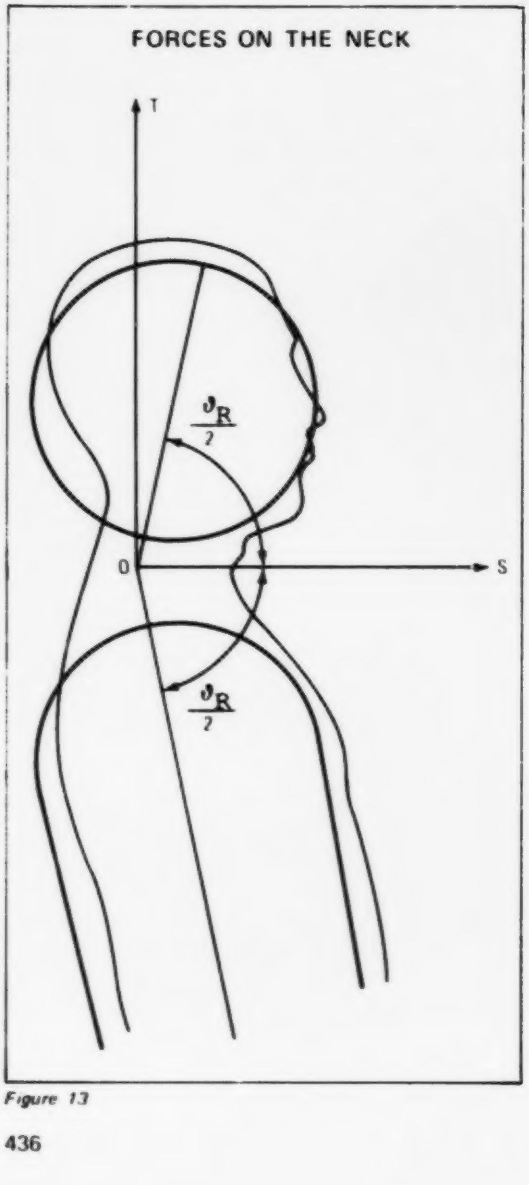

Figure 13

the hinge, and evaluated in the direction normal to S (Figure 13).

Figure 16 — Force (in Kg) applied by the curtain to the head.

In all cases, the neck and head stress reduction due to the head restraint system is very noticeable.

In the table contained in Figure 17, we have summarized the results obtained in the two cases, by comparing the maximim absolute values of the above described quantities.

Conclusions

It was possible, for both cases examined, to keep the torso decelerations below 60 g for a frontal crash at 40 mph, assuming 550 mm as the useful crushing distance of the front of the vehicle (about 900 mm would have been required for a 50 mph crash).

The calculated stresses in the head and neck of the passenger with safety belts but without a head restraint device (Case 1) were extremely high. Although the reliability of the results obtained in this case is lessened by the doubts expressed earlier on concerning the dynamic behavior of the neck in the hyperflection zone, nevertheless the order of the calculated stresses indicates that very probably the

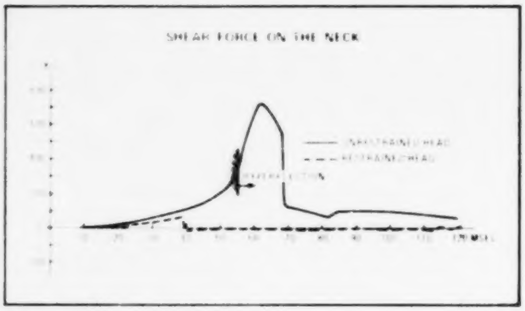

Figure 14

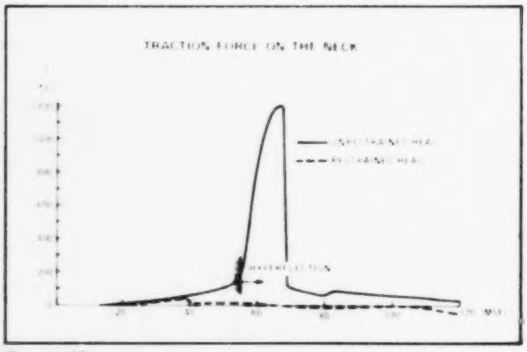

Figure 15

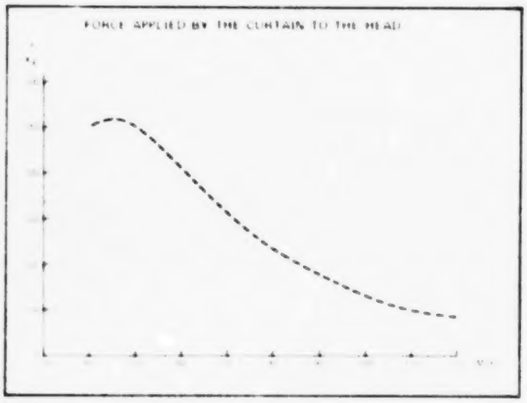

Figure 16

physiological endurance limits for the head and neck will be overreached.

The calculated stresses in the head and neck of the passenger with safety belts and also with the head restraint device considered (Case 2) were very low, and certainly under the physiological endurance limits. It must, however, be remembered that these results were obtained assuming a critically damped, elastic head restraint system, which may be rather difficult to achieve in practice. Elastic systems having lower damping effects (hence easier to make) will give less favorable, but still very interesting results.

	SUMMARY						
	HEAD				NECK		TORSO
	g	rad/s²	SI	F kg	S kg	T kg	g
UNRESTRAINED HEAD	135	10800	2650		730	1200	53.5
RESTRAINED HEAD	45	1200	450	260	60	40	48

Figure 17

THE ITALIAN TECHNICAL PRESENTATION

SECTION TWO

PART 7

EASY BELT – SOFT SALLET SEAT

CARLO BIANCHI ANDERLONI
Presented by
ANGELO SCHIEPPATI, *Senior Active Safety Engineer*
Alfa Romeo S.P.A.

We would like to discuss one of the safety devices developed by us as part of our work in the ESV field. It consists of an occupant restraint system using sophisticated passive belts, working in conjunction with a head restraint device.

Starting with the assumption that the only important result to be achieved is to save human lives, then it seemed obvious to us that there is no point in imposing a given shape and maximum value for the passenger compartment deceleration wave, or even specifying a maximum or minimum crushing distance for vehicles undergoing ESV type crash tests. What is important for occupant safety is a limited compartment deformation, the maximum possible deceleration distance and optimized restraining systems; the combination of these factors will result in the lowest obtainable forces and stresses on the occupants.

Therefore, in the following discussion, we will ignore those ESV requirements regarding the vehicle crushing distance and the characteristics of the passenger compartment deceleration wave. Instead, we will assume that the problem of limited compartment deformation which is a structural question, is outside the scope of this part of our work. Incidentally, our paper presented at the Third International ESV Conference in Washington last year proved that the law of motion of the restraint anchorage points has little bearing on the motion of the occupant. Therefore, we will deal here only with what happens to the occupant itself.

Faced then with the problem of developing a passive restraint system which should, at the same time, be reasonably uncomplicated, offer adequate protection in the greatest possible number of crash modes, and present a favorable balance between advantages and disadvantages, we had to choose between air bags, belts and systems involving only fixed paddings. These latter systems were discarded because their bulk is totally incompatible with the size available in our vehicles, and also because of the adverse psychological effects (claustrophobia) they are known to provoke on the occupants.

We have, for the moment, discarded air bags, after a considerable amount of testing and research, because of their well known disadvantages, even if we certainly acknowledge their undoubted efficiency in frontal crashes.

Among the known restraint systems, we were then left with safety belts. We therefore tried to improve on the present belt systems, both from the point of view of occupant protection and also ease of wearing.

EASY BELT

We thought that an improvement on the traditional 3-point system would be one derived from the well known full harness type; that is, a system consisting of a lap belt, together with two symmetrical chest restraints.

The disadvantages of the traditional system on the symmetrical "4-point" one are well known:

a. Smaller chest contact area, and hence higher pressures.
b. The forces on the chest are not distributed symmetrically.
c. In case of a frontal crash, the body is badly controlled in its foreward motion, along a trajectory of uncertain definition. Above all, there is the well-known dangerous rotation about the spine due to restraining only one shoulder, as can often be noted in crash tests.
d. When the crashes are not frontal, the body is badly restrained.
e. An incorrectly positioned passenger will not be brought to a proper posture in case of a crash.
f. The possibility of strangulation caused by the diagonal belt is high.
g. There is less adaptability of the system to all occupant percentiles, especially to children.

The superiority of full harness belts is further accentuated by the fact that they are used by aircraft pilots and racing car drivers, where the chances of a high speed accident are far greater than for an every-day driver.

In fact, Alfa Romeo's experience in racing

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throughout the world has further strengthened our belief in the advisability of this type of restraining system. On the other hand, all these unquestionable advantages are offset by the difficulties of putting on the belt when sitting in the car.

The solution, then, is to enable the occupant to use his belt easily and, if possible, automatically. We think that we have successfully solved these problems with our "Easy Belt" system, which requires a minimum of effort by the occupant.

The system consists of two moving arms, each with one end hinged about halfway up the seat back. At the other extremities of these arms, lap belt retractors are fitted; the ends of the shoulder belts are fixed to the lap belt (Figure 1). The other ends of the shoulder belts are wound on an emergency locking retractor placed under the seat. This emergency locking retractor normally enables the occupant to move freely, but locks immediately when a predetermined deceleration is reached (Figure 2).

As shown in Figure 3 the only effort required by the occupant as the belt descends (in the case of an automated system) is for him to raise his arms, to enable the lap belt to come properly into position (Figure 4). To free the belts, the occupant has only to release one of the latches, and the arms move automatically, or manually, back into the vertical position.

Occupant access into the car is not in any way impeded; the arms are upright and the lap belt rests against the roof. When the occupant sits on his seat, the arms move forward and downward (either automatically or manually) carrying with them the lap belt, which then latches onto a mechanism mounted on either side of the seat. This ensures the correct positioning of the belt and provides a means of reacting to the loads imposed on the belts during a crash (Figure 5).

The retractors maintain the correct belt tension on the occupant and at the same time automatically adjust the belts to fit occupants of any percentile. In case of accidents the retractors lock and all the loads are taken by the seat.

SOFT SALLET

As we said before, the system consists also of a head restraint device. This is an indispensable complement to a belt system, if it is wished to raise the crash speed at which the protective system is effective. The basic idea is to restrain the head, in its movement relative to the torso, by means of a woven curtain which would automatically position itself in front of the occupant's head in case of an accident; the head would then be decelerated and guided in its forward motion by the tension forces in the curtain, the energy being absorbed by the elongation of the curtain material.

By varying the rigidity of the curtain material to suit the vehicle deceleration characteristics, the full harness belt elongation, and its energy absorption properties, the system can be adjusted to give acceptable head decelerations and rotational movement relative to the torso.

The deployment of the head restraint may be obtained by inflating the whole curtain, or only part of it. Such deployment may be controlled by an inertial sensor fixed to the vehicle, or by a device

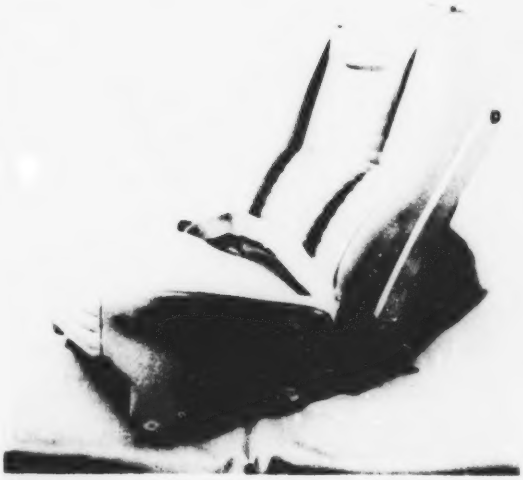

Figure 1

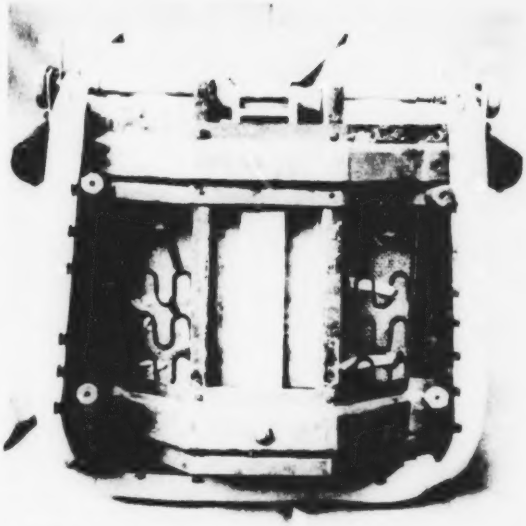

Figure 2

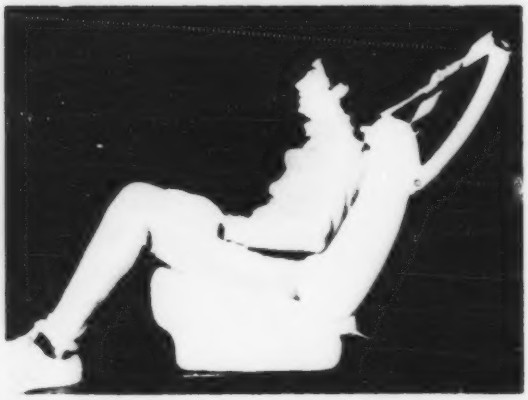

Figure 3

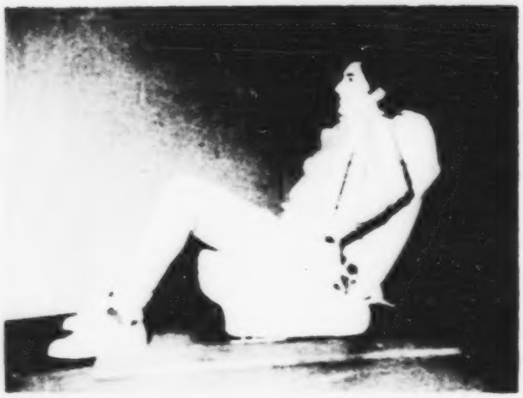

Figure 4

sensing a pre-determined tension in the belts. Whatever system will be used, the volumes to be inflated will be considerably smaller than those of air bags, with resulting lower noise levels.

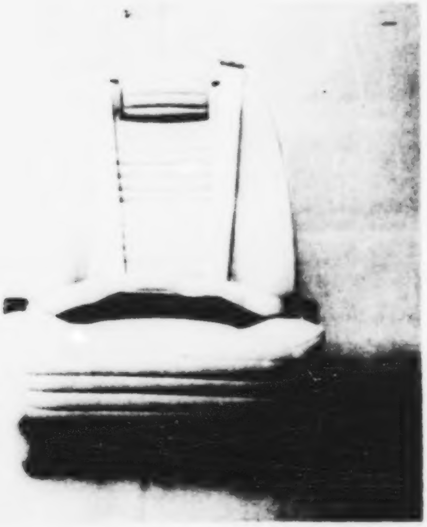

Figure 5

We have studied two different head restraints based on these principles. The first consists of a curtain which, in its rest position, is folded against the roof, and is supported by a light frame carrying no load. A crash signal by the sensor will cause the downward deployment of the curtain; the loads imposed on the system while it restrains the head will be taken by suitable curtain anchorage points above the seat back.

The second solution consists of a curtain which, in its rest position, is folded against the shoulder belts and the lap belt; the curtain deploys upward, restraining the occupant's head.

At the end of the paper we have circulated, you will find the mathematical definition of the principles involved.

THE ITALIAN TECHNICAL PRESENTATION

SECTION **TWO**

PART **7**

PROGRESS REPORT OF THE ESV DEVELOPMENT AT FIAT

VITTORIO MONTANARI, *Director*
Applied Research
Department of Motor Cars
Fiat S.P.A.

INTRODUCTION

During the nine months which have elapsed from the Washington Conference to this Kyoto Conference, Fiat has accomplished the program started in the spring of 1971.

In detail:

- Research work on the structure and compatibility of the first 1,500 pound car has continued.
- Investigations were started on the development of restraint systems for the 1,500 lb. car.
- Two 2,000 pound and 2,500 pound car prototypes were built and an intense test program has begun which, for the time being, is limited to the initial stage; namely, investigations on structure behavior.
- Theoretical and experimental research was carried out on the vehicle handling problem.
- Finally, there has been considerable research activity on the problem of compatibility: the objective of this phase is to build up the most complete picture possible of the situation in order to highlight the different aspects of this complex problem.

The Fiat technical presentation consists of the following parts:

- Fiat analysis of road accidents in Italy
- Some considerations on vehicle handling and braking performance
- Research on the 1,500 pound – 2,000 pound – 2,500 pound ESV
- Fiat ESV tests
- Research on restraint systems
- Compatibility

The following persons have assisted Mr. Vittorio Montanari – Fiat ESV Program Project Manager – in preparing this technical presentation:

- Mr. Giampaolo Boano – In charge of the Styling Center; responsible for styling.
- Mr. Emilio Deaglio – Responsible for coordination of prototype manufacture.
- Mr. Enzo Franchini – In charge of the Safety Laboratory; responsible for vehicle tests.
- Mr. Alfredo Margara – In charge of the Publication and Standards Dept.; responsible for statistics and coordination of the technical presentation
- Mr. Giuseppe Puleo – In charge of the advanced Car Design Department; responsible for ESV design.
- Mr. Riccardo Scalet – In charge of the Calculations Dept.; responsible for mathematical support and computations.
- Mr. Luigi Zandonà – In charge of the vehicle Development and Test Laboratory; responsible for road tests.

FIAT ANALYSIS OF ROAD ACCIDENTS IN ITALY

At the third ESV Conference in Washington, we announced that we had started a systematic analysis of road accidents in which Fiat-built cars were involved. The investigation is being carried out in 13 Accident Investigation Centers which are distributed over an area accounting for 40% of the total number of cars and 50% of the total number of accidents.

At the May 1972 ESV Conference in Washington we presented the results of 410 accidents. The investigation was carried on at a rate of about 200 cases a month, reaching the present total of 2,200 cases. The analysis includes a diversity of models, which are representative of the entire range of Italian cars, and is limited to Fiat-built cars (which account for about 70% of the total number of cars on the road, making them a representative sample).

Figure 1 shows the percentage distribution values submitted in Washington last year against the present values, which take into account the models subsequently put on the market (i.e. the 126 – which is included in the "500" class – and the 132, in the "125" class). Distribution has not changed substantially; an increase is recorded for the "128" class, which is gaining wider popularity.

Accident distribution is shown in Figure 2 and refers to a total of 410 cases (Washington) and to a

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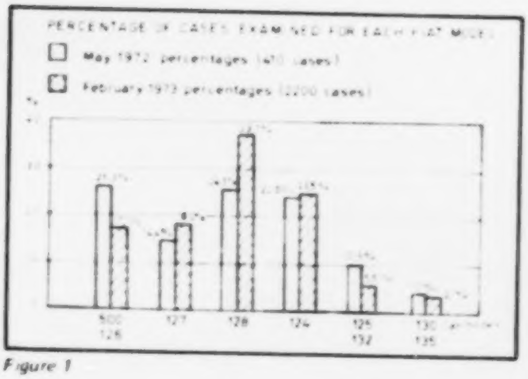


Figure 1

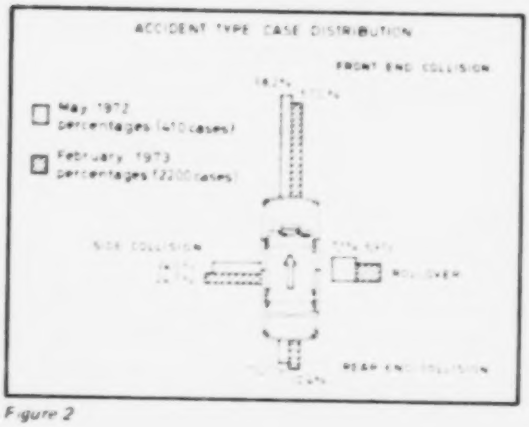


Figure 2

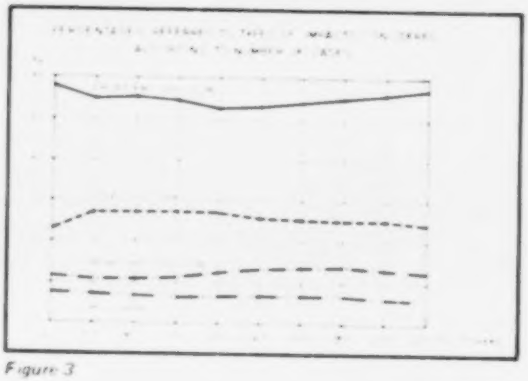


Figure 3

total of 2,200 cases (Kyoto), according to collision type: front end, rear end, side and rollover. Percentages remained substantially unchanged even if the number of investigated cases was progressively increasing (Figure 3).

Distribution, based on the severity level of damage to the car, (Figure 4) was plotted according to the five severity levels established by the Cornell Aeronautical Laboratory ACIR scale. In this case, too, value fluctuation, as a function of the increasing number of investigated accidents, is quite low (Figure 5). It appears that the percentage of moderate damages has gone up while that for more severe damage has gone down.

The distribution of occupant injury severity levels is plotted over five classes: uninjured, contused, lightly injured, seriously injured, fatalities (Figure 6).

Fluctuations are contained, as shown in Figure 7; a reduction in fatality percentage is recorded, along with an increase in the "uninjured" percentage. It is our intention to extend this investigation further by bringing the number of cases investigated up to several thousands, so as to facilitate statistical processing on a computer, with the aim of obtaining more detailed data about the different characteristic parameters.

Furthermore, we are working in cooperation with the other members of the Committee of Common Market Manufacturers in order to obtain joint data of European validity.

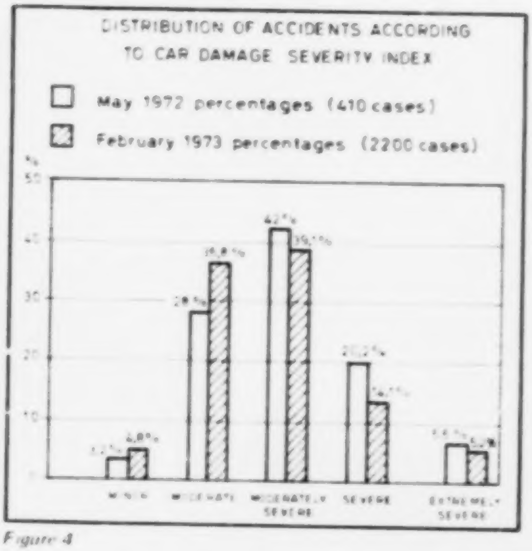


Figure 4

However, we can point out that the modest fluctuations recorded with a case increase from 410 to 2,200 are substantially a confirmation of the first conclusions which Fiat presented at the Washington Conference, some of which are as follows:

- The most frequent type of accident is the front-end collision followed, in decreasing order of frequency, by side collision, rear-end collision and rollover.

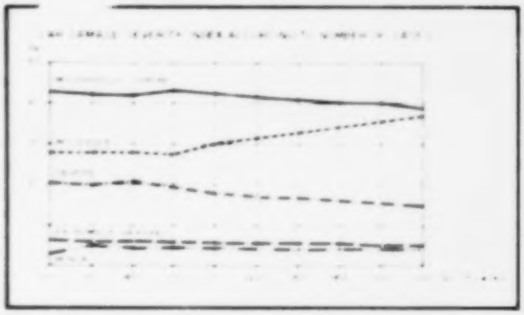


Figure 5

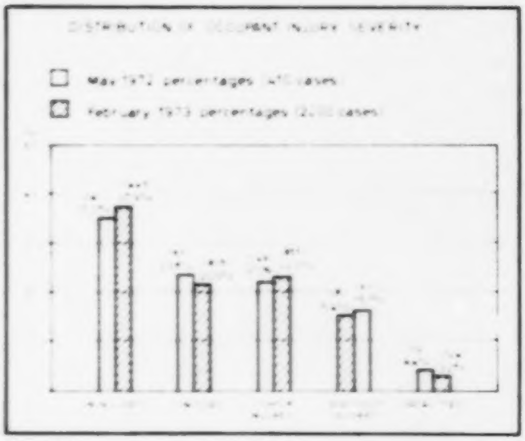


Figure 6

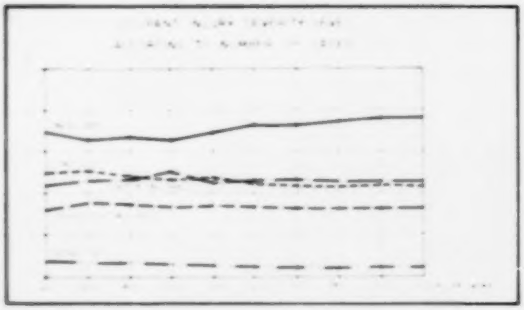


Figure 7

- 80% of the accidents cause, on average, moderate to severe damage to cars.
- 80% of the accidents cause, on average, light injuries to occupants.

SOME CONSIDERATIONS ON VEHICLE HANDLING AND BRAKING

Systematic research was started at Fiat some years ago to define objectively what should be meant by "safe vehicle" from the standpoint of handling and braking performance. Experimental verifications and computer simulations using a complex mathematical model of the vehicle are used in this work.

Research is still in course; therefore, it is not our intention to present conclusions here; we only wish to examine some proposals of the U. S. ESV Specification and to submit some other evaluation parameters and test methods now under study at Fiat.

The subjects we shall be dealing with are:

- Steady-state response
- Transient yaw response
- Releasing the accelerator from steady state
- Rapid lane changing
- Braking performance

Steady-State Response

Earlier U.S. ESV Specifications required the evaluation of a vehicle under steady state, observing the pattern of yaw velocity to steer angle ratio as a function of speed for a .4 g lateral acceleration.

The value of this ratio depends on the wheelbase of the vehicle being evaluated. In the latest NHTSA Specification for the intermediate ESV, this evaluation index has been multiplied by the wheelbase in order to partially eliminate this incongruence.

The chart in Figure 8 shows the response curves plotted from test results and calculated data for two Fiat production cars, and evaluated according to the required method.

Note how all curves are concaved downward, first increasing and then decreasing. We think this pattern is characteristic of all understeering vehicles.

To demonstrate this, let us consider an extremely simplified model of the vehicle. Track and wheelbase are negligible with respect to the circular path radius; the slip angles are a function of lateral acceleration alone and not of velocity. Figure 9 shows the equation which relates to this model.

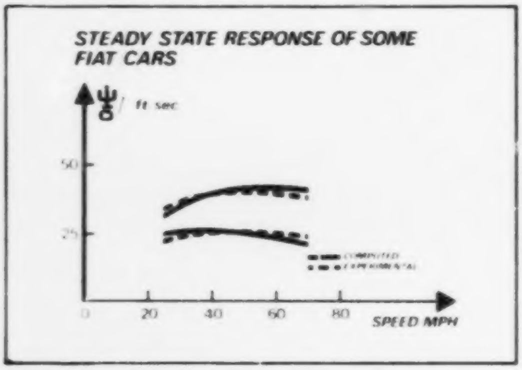


Figure 8

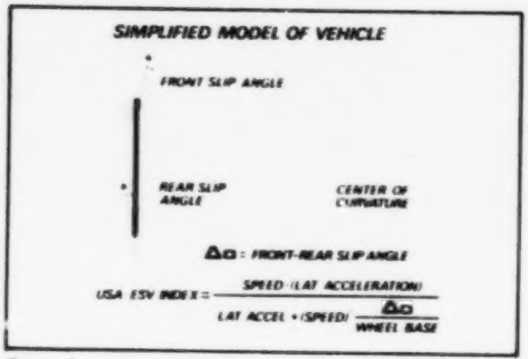

SIMPLIFIED MODEL OF VEHICLE
FRONT SLIP ANGLE
REAR SLIP ANGLE
CENTER OF CURVATURE
Δα : FRONT-REAR SLIP ANGLE
USA ESV INDEX = SPEED (LAT ACCELERATION) / LAT ACCEL + (SPEED) Δα / WHEEL BASE

Figure 9

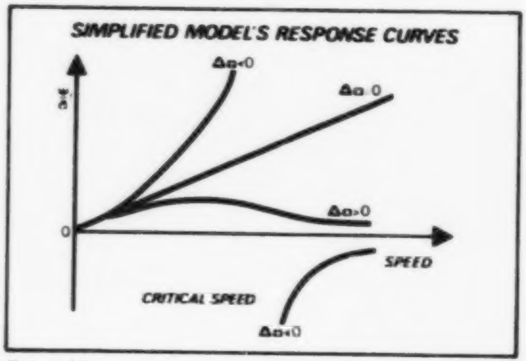


Figure 10

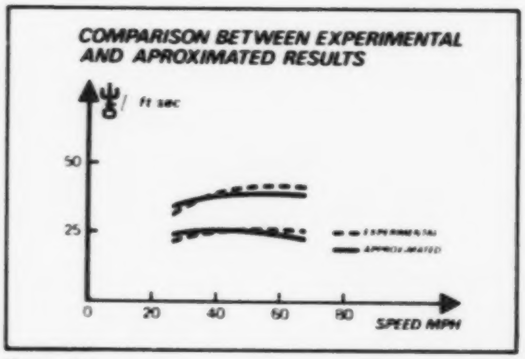


Figure 11

For a given wheelbase and acceleration, the parameter that individuates the required ratio pattern is $\Delta\alpha$. Figure 10 illustrates this pattern.

- For $\Delta\alpha = 0$ (central vehicle) the response is a straight line, known as neutral line.
- For $\Delta\alpha < 0$ (oversteering vehicle) the response is a curve above the neutral line up to a *critical speed*, beyond which the steer angle becomes negative, and the curve runs below the speed line.
- For $\Delta\alpha > 0$ (understeering vehicle) the response curve reaches a maximum and then tends to zero.

Let us now compare the experimental curves of two cars with the approximate curves calculated by the simplified formula. The similarity of the curves compared (Figure 11) indicates that the determination of the vehicle response at different speeds is not of great interest, as the pattern obtained with the simplified model (entirely characterized by parameter $\Delta\alpha$) is satisfactory. In other words, if one point is known, the whole curve is known.

What has been said so far brings out the importance of parameter "$\Delta\alpha$", that is the difference between the front and rear slip angles, in characterizing vehicle behavior at a given lateral acceleration. We think that this, or another equivalent parameter, can be taken as the behavior index of the vehicle under "steady-state" conditions. A parameter equivalent to $\Delta\alpha$, but more easily measured, is $\Delta\delta$ which is the difference between the actual steer angle and the Ackermann steer angle. The latter can be assumed as equal to the steer angle on the same radius, travelled at very low speed. The equality $\Delta\delta = \Delta\alpha$ holds good for the simplified vehicle.

The foregoing facilitates the extension of the analysis to lateral accelerations other than .4 g. This extension is indispensable because the behavior of a vehicle can change substantially as lateral acceleration varies.

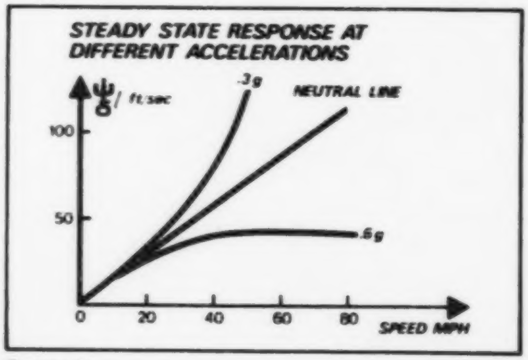


Figure 12

Referring to Figure 12, note the different response curves of a Fiat prototype, determined at lateral accelerations of .3 g and .6 g respectively.

The chart in Figure 13 shows, for the same prototype, the pattern of parameter "$\Delta\delta$" as a function of lateral acceleration. It is clear that this method of representation describes vehicle behavior synthetically showing how oversteer changes to understeer. We think that much information can be derived from the study of the function represented in this manner: for instance, many steering indexes so

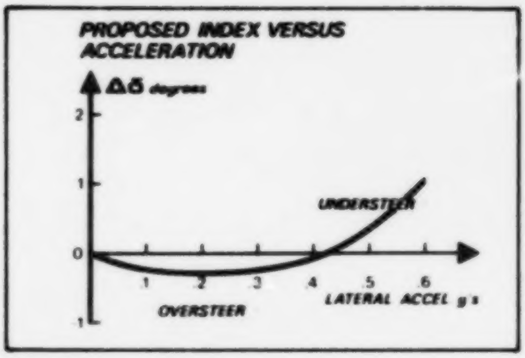


Figure 13

PROPOSED BY	DEFINITION	CORRELATION
USA ESV	$i = \frac{\dot{\psi}}{\delta} \cdot l$	$i = \frac{V \cdot a}{a + V^2 \frac{(\Delta\delta)}{l}}$
UNIVERSITY OF MICHIGAN	$i = \frac{\dot{\psi} \cdot l}{\delta \cdot V}$	$i = \frac{1}{1 + \frac{V^2 \cdot \Delta\delta}{l \cdot a}}$
FORD	$i = \frac{1}{R^2} \cdot \frac{\partial R}{\partial a}$	$i = \frac{1}{l} \cdot \frac{d(\Delta\delta)}{da}$
SAE	$i = \frac{V^2}{l} \cdot \frac{\partial \delta}{\partial a}$	$i = 1 + \frac{V^2}{l} \cdot \frac{d(\Delta\delta)}{da}$
MIRA	$i = \frac{\frac{\partial \dot{\psi}}{\partial V}}{\frac{\partial(\delta \cdot V)}{\partial V}} \cdot l$	$i = \frac{1}{1 + \frac{R}{l} \cdot \Delta\delta} \cdot \frac{1 + \frac{V^2}{l} \cdot \frac{d(\Delta\delta)}{da}}{1 + \frac{V^2}{l} \cdot \frac{d(\Delta\delta)}{da}}$
MIRA	$i = \frac{\frac{\partial \dot{\psi}}{\partial \delta}}{\frac{\partial(\delta \cdot V)}{\partial \delta}} \cdot l$	$i = \frac{1}{1 + \frac{V^2}{l} \cdot \frac{d(\Delta\delta)}{da}}$

NOTATION

δ STEER ANGLE
$\dot{\psi}$ YAW VELOCITY
l WHEELBASE
a LATERAL ACCELERATION
R RADIUS OF STEERING PAD
i INDEX
V SPEED
$\Delta\delta$ PROPOSED INDEX

Figure 14

far proposed can be obtained from it and its derivative through simple analytical formulas (Figure 14).

An advantage of the study of the behavior of a vehicle under "steady-state" conditions by means of the proposed index is that it is sufficient to take one radius of the steering-pad, varying the speed only.

The chart in Figure 15 illustrates the experimental results obtained with a Fiat vehicle for different radii. Should the slight scatter of results be confirmed in the future, it would prove that Δa is substantially independent of speed.

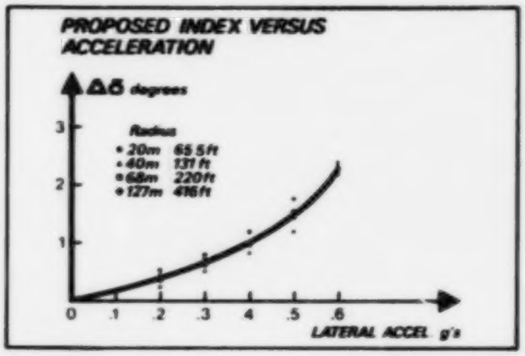


Figure 15

Transient Yaw Response

The test proposed by the U. S. ESV Specification requires the evaluation of the yaw velocity to steer angle ratio versus time. After the first steering input instants, the steer angle remains constant. Therefore, the evaluation required by the test is, in practice, yaw velocity pattern in time.

In our opinion, the objective of this test is the evaluation of two characteristics of the vehicle:

- Transient stability, with the 70 mph test.
- Response rapidity, with the 25 mph test.

For *transient-stability* evaluation we feel that it is useful, but not sufficient, to know the yaw velocity pattern. For instance, we think it is necessary to take into account sideslip velocity which, together with yaw velocity, determines the centrifugal acceleration.

Figure 16 shows the calculated response curves of a Fiat car at 25 mph. During transient motion, ratio ρ

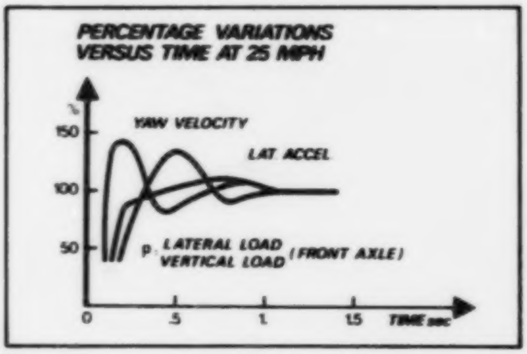


Figure 16

should not greatly exceed the steady-state value to which it tends. Magnitudes are expressed as percentages, with respect to the steady-state value. Note how the patterns differ. The curves of the same car at 70 mph are shown in Figure 17. Compared with the curves at 25 mph their pattern is completely different.

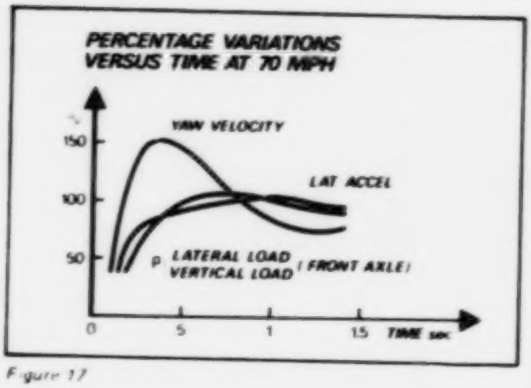


Figure 17

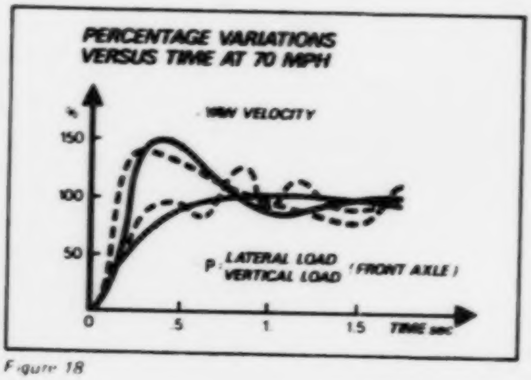


Figure 18

The last chart (Figure 18) shows a comparison between the previous car (solid line) and the same car without shock absorbers (broken line). Note how this important modification is evidenced not so much by the yaw velocity but, rather, by the adhesion limit factor (ratio ρ). We are not going to suggest any conclusions at this point. We only wish to highlight the need not to restrict the investigation to yaw velocity alone.

To evaluate *vehicle responsiveness* seen as the ability to change direction readily, we feel that course deviation is of primary importance. It depends on two factors

- Yaw velocity
- Sideslip velocity

When the latter attains significant values, discrepancies arise between U. S. ESV Specification and course deviation.

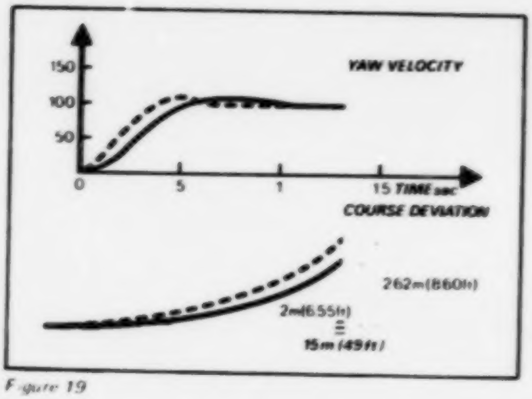


Figure 19

The response curves and the courses of two different vehicles submitted to the "transient-yaw" test given in the U. S. ESV Specification are shown in Figure 19. It should be noted that course deviation draws sufficiently clear distinctions.

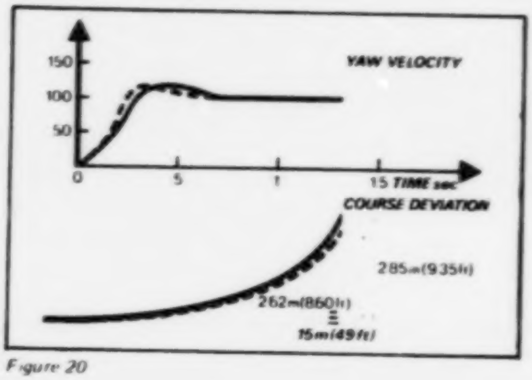


Figure 20

Figure 20 shows how a car, rated as more responsive than another on the basis of yaw velocity (broken line), is, in practice, less responsive when rated versus course.

Figure 21 shows how cars, rated differently on the basis of yaw velocity, prove practically identical when rated versus course.

For "transient-yaw response", as for steady-state, we feel that to limit the investigation to just one stabilization acceleration is not sufficient. Shown in Figure 22 are the paths of a vehicle which moves onto the same stabilization radius at two different speeds which correspond to steady-state lateral accelerations of .2g and .6g respectively. For the same vehicle the chart of Figure 23 shows the family of paths at different stabilization accelerations.

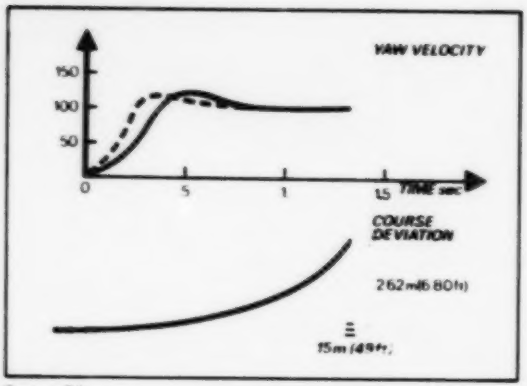


Figure 21

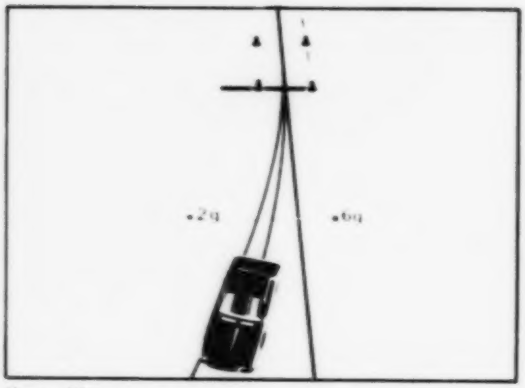


Figure 22

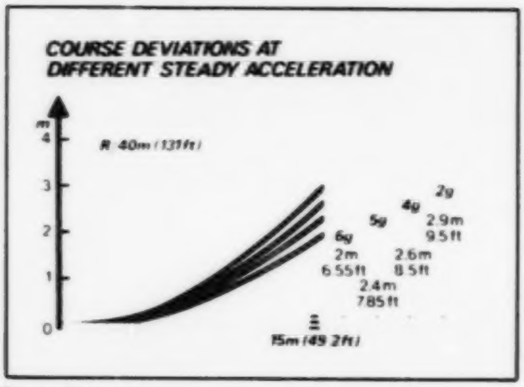


Figure 23

Releasing The Accelerator From Steady-State

In addition to the previous maneuver (which represents a transient consequent to a constant traction steering input) we feel it advisable to examine a transient consequent to a constant steer traction input. That is, the accelerator pedal is released starting from steady state on steering-pad and with steering wheel locked.

The desired reduction in speed is accompanied by a deviation from the initial path. With reference to this deviation, we think that car behavior can be rated good if deviation is slight or bad if it is large.

The path response of the vehicle depends on initial lateral acceleration. Shown in Figure 24 are the paths followed by the same car on which accelerator release is started (the radius being always the same) at accelerations of .2 g and .6 g respectively. It is apparent that the path is progressively closing in as speed (and hence acceleration) increases.

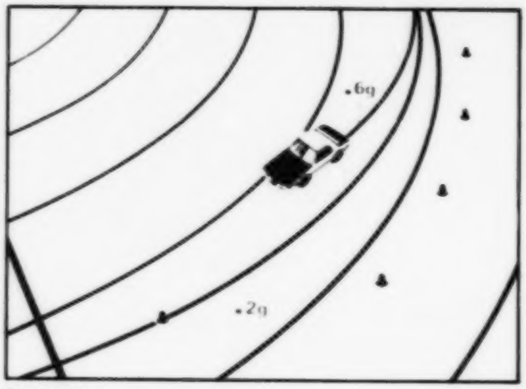


Figure 24

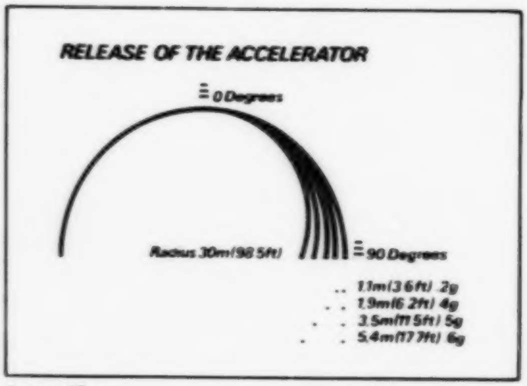


Figure 25

The chart in Figure 25 provides the family of paths for different initial accelerations for the same vehicle. To study one aspect of this maneuver, let us examine Figure 26 where the circular path radius variation as a function of time is shown for the same vehicle given in the previous chart. As at the end of the maneuver the cars will be "almost stopped," the asymptote to which curves tend is Ackermann's curvature radius, which for understeering vehicles will be shorter than at the beginning. The asymptote value depends on vehicle behavior under steady-state con-

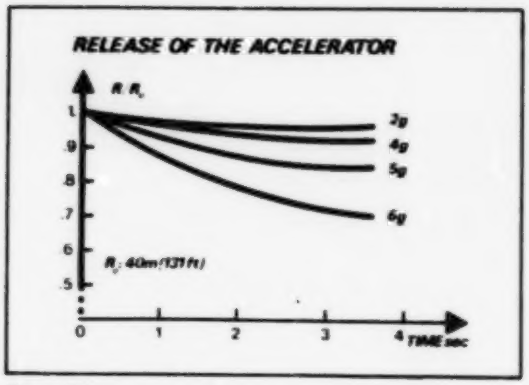


Figure 26

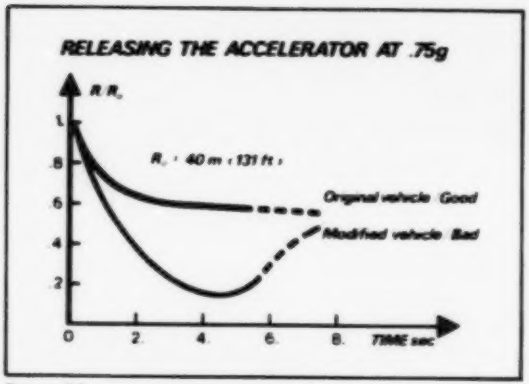


Figure 27

ditions; the shape of the curve depends on transient behavior.

Figure 27 shows how the modifications introduced in the prototype are evidenced by the accelerator release test. We think that, to be qualitatively acceptable, a vehicle must give a response whose curve should never intersect the asymptote and should not depart too quickly from the initial value.

The research underway aims at establishing an organic group of tests run on the basis of only one stabilization radius and at different lateral accelerations. This approach, if found valid, would smooth out the researcher's task, since no large testing grounds would be required. In any case, we feel it would give a more comprehensive picture of vehicle behavior than a group of tests based on only one lateral acceleration.

Rapid Lane Changing

The rapid-lane-changing test is representative of an emergency maneuver which is very frequent in traffic to avoid a sudden obstacle. It is felt that this test is highly significant in evaluating car performance. As a result, the problem was tackled with a general approach by examining two test procedures fixed path and imposed maneuver.

1. The fixed-path test was run with driver and consisted of driving the car along the course shown in Figure 28 at the maximum possible speed, while allowing the driver freedom to use the steering wheel and the accelerator as he deemed best.
2. The imposed maneuver test (with sinusoidal steering input) was planned to be run with driver and automatic control; the vehicle, travelling at a set speed, was submitted to a steering input following a sinusoidal law with a given cycle and a progressively larger amplitude (see a typical recording in Figure 29).

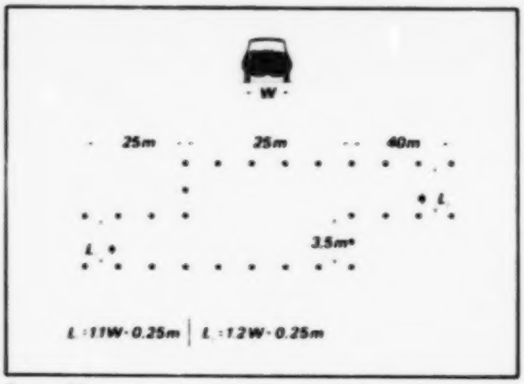


Figure 28

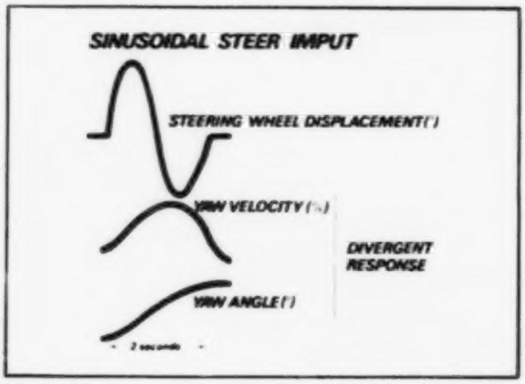


Figure 29

In both cases, the parameters of driver behavior and car dynamic were read off on-board transducers.

During the fixed-path test the results recorded with two drivers of comparable skill evidenced that, with the same maneuvering time, the steering control laws were quite different, as may be seen from the graphs shown in Figure 30.

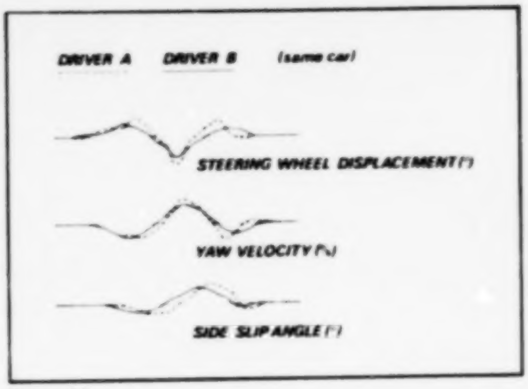


Figure 30

We now have an investigation underway to see if it is possible to find an index which is representative of car behavior even though the courses possible in the fixed path chosen may be different.

As regards the imposed maneuver, we are now only at the testing stage with driver. On the basis of the recorded data, we are trying to correlate this information with the impression of the driver, especially when, at the wheel of the off-design car, he judges that it is no longer acceptable. We are not in a position to submit results or conclusions on this work, because we feel we should test a larger number of cars and learn to interpret the data better.

What is really meant here is that we think these test methods are highly significant for the evaluation of accident avoidance on the road, especially if we succeed in correlating their results with those obtained from the tests described elsewhere in this presentation.

Braking Performance

There are two basic factors in the evaluation of brake systems in the light of active safety: braking effectiveness and vehicle directional stability in braking.

A number of tests have been run by Fiat to assess the behavior of a car under braking conditions, and we believe an adequate evaluation procedure has now been developed. According to this, the braking performance of a car can best be evaluated by running straight-line and in-a-turn braking tests over surfaces with different friction coefficients, and by using cars with different loads.

The tests now being conducted are aimed at highlighting any significant parameters that may pick out good axle braking distribution and reveal car directional instabilities, if any. In all our tests the steering wheel is locked in the brake application position, because the remedial steering inputs by the driver of a skidding car do not always yield satisfactory results.

In the straight-line test, the braking performance is determined at the instant of incipient locking of any one wheel, because it is quite clear that the locking of a wheel may cause a condition of instability and danger.

Braking efficiency can thus be defined as:

$$\eta = \frac{Ax}{\mu}$$

where Ax is the deceleration value expressed in g's obtainable at incipient locking limit of any wheel, and μ is the maximum value of the test surface friction coefficient. As such, it is a measure for evaluating the capability of a car to best exploit the adhesion limit available during brake activation.

Since in a turn-braking test wheel locking is followed by a quick variation in yaw velocity it follows that vehicle yaw velocity must be recorded continuously. The same braking efficiency parameter defined earlier, is also taken into consideration in turn braking, except that, here, by parameter we mean the maximum braking effectiveness obtained in a turn (under safe condition) in respect to the initial lateral acceleration.

The information needed for the evaluation of car behavior in the two braking modes defined above is obtained by on-board sensors (one is a stabilized inertial platform) (Figure 31), and a radio monitoring system with an on-ground autonomous portable receiving station capable of both analog and digital recordings.

Noteworthy is that by using the above instrumentation the car can be judged without ground references because the parameters can be read off directly and no calculation is needed.

The parameters recorded are:

- Vehicle velocity
- Brake master cylinder output pressure
- Wheel angular velocities, both front and rear
- Longitudinal deceleration
- Yaw velocity

Following are some results of tests conducted on cars with brake systems in standard and in deliberate off-design condition (the latter has the purpose of obtaining unacceptable behavior).

Straight-Line Braking Test

The straight-line braking test began when the car was driven at 40 mph in a 12 foot wide lane. With the steering wheel locked, the driver slammed the pedal

Figure 31

against an adjustable stop. The line pressure remained constant throughout the braking phase.

The response of a front axle biased car (i.e. one on which braking is more powerful on front axle) to the surface friction coefficient is shown by the typical recording in Figure 32. In this type of braking-power distribution, the front wheels were obviously subjected to a considerable degree of slip over the ground. As shown, front wheel slip was always higher than rear wheel slip.

Rear wheels retained effective directional stability, vehicle path was maintained right up to the time the car was coming to a stop. Zero yaw velocity was evenly maintained at the instant of incipient wheel locking.

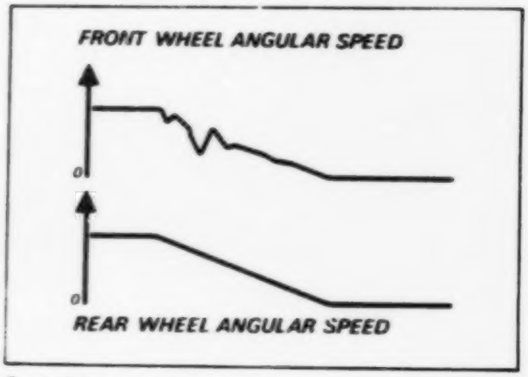


Figure 32

Let us now turn to the opposite type of braking power distribution, (i.e. the response of a rear axle biased car with more powerful braking on rear axle).

On application of the brakes slip values were reversed (Figure 33), and zero yaw velocity was no longer maintained (Figure 34) at the instant of incipient locking of either rear axle wheel, because the rear axle was then incapable of maintaining directional stability. Under such conditions, incipient wheel locking may give rise to spin-out.

The pressure-deceleration chart (Figure 35) shows the braking efficiency of two cars; the front axle biased one is characterized by high braking efficiency (curve b), whereas the rear-axle biased one shows a lower efficiency (curve a).

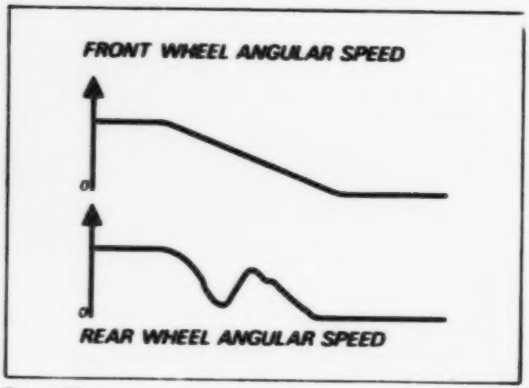


Figure 33

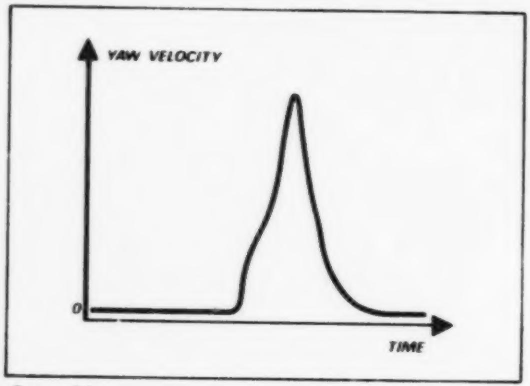


Figure 34

Braking In A Turn

Brakes were applied when the car was moving at 40 mph steady state and .3 g lateral acceleration. Just as in straight-line testing, the driver slammed the

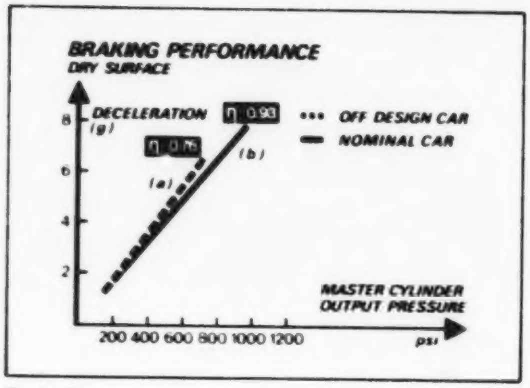


Figure 35

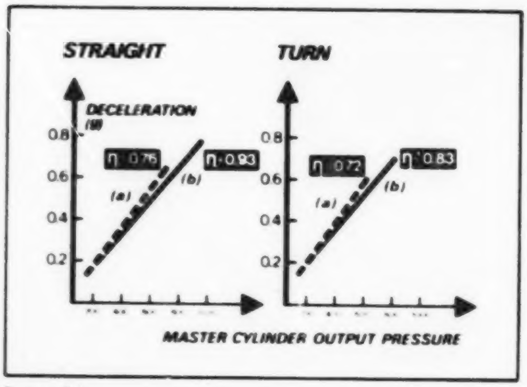


Figure 36

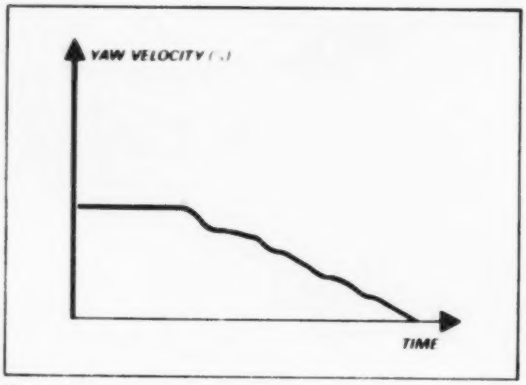


Figure 37

brake pedal to obtain the required line pressure, the steering wheel was locked. The braking performance parameter remained significant also under these conditions. A comparison can therefore be made with straight-line test results (Figure 36).

The difference in braking efficiency between front (a) and rear (b) biased cars on a straight-line test is shown in the left hand side of the chart; whereas that in braking efficiency in a turn between the same two cars is reported on the right. Obviously, braking in a turn calls for greater care because part of the available tire-ground interface adhesion goes into obtaining the reaction to offset the centrifugal force.

In incipient wheel locking the curvature covered by the front biased car was smaller than that of the original trajectory, whereas yaw velocity decreased steadily (Figure 37).

The rear biased car went into an incontrollable skid, while yaw velocity reached its peak (Figure 38).

The foregoing considerations should be taken into account when making an overall assessment of acceptable braking performance levels; in particular, it should be noted that the higher the braking effectiveness, the better off the car is, providing no increase in yaw velocity is recorded when brakes are applied at the limit of incipient wheel locking.

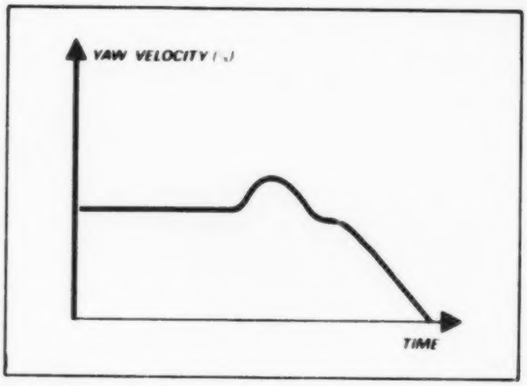


Figure 38

RESEARCH ON THE 1,500 POUND – 2,000 POUND – 2,500 POUND EXPERIMENTAL SAFETY VEHICLES

The 1,500 Pound ESV (Figures 39, 40 and 41)

After presentation at the Third ESV Conference in Washington of a first prototype and test results on the small 1,500 pound ESV, its design and experimental development work was carried on without interruptions or delays.

The number of prototypes built to date is 13, that is, three more than the initially set total for completion of the basic research on passive safety problems. This means that up to 11 prototypes were built during the nine months since the last technical conference. Of these, 10 have already been used in destructive tests and one was kept for this presentation. Noteworthy is that one prototype was placed at the disposal of the U. S. Department of Transportation for collision tests against the A.M.F. 4,000 pound ESV prototype under the research program on compatibility problems in crashes between vehicles of different mass and structure.

2nd PHASE OBJECTIVES

1 - **Passing the 50 MPH front end collision test against pole**
2 - **Manufacture and testing of a lightened prototype**
3 - **Improvement of the low speed protection system**
4 - **Installation and testing of restraint systems**

As indicated above the first objective was the completion of the basic collision tests required by the ESV Specification, namely, the verification of the vehicle structure crashworthiness in a front end collision test against a pole at 50 mph. As may be recalled, no results on the test could be submitted in Washington because of the unavailability, in useful time, of a prototype other than the one which had already been submitted to as many as four severe tests: 50 mph front and rear end barrier collision, 15 mph side collision against pole and drop test.

The satisfactory behavior of the small ESV during these first tests had led us to believe that it would do just as well in the front end pole test. And this, notwithstanding the fact that earlier experience on suitably reinforced standard production cars had evidenced the particular severity of this type of test. Unfortunately, our hopes were bound for disappointment after undergoing the test in question, prototype No. 3 revealed unexpected deficiencies in the strength of some parts of the front structure, with consequent unacceptable passenger compartment intrusion values.

It then became necessary to introduce some structural modifications and build an updated prototype for repetition of the test. The provisions adopted proved effective and the second trial was successful. However, another item had to be entered in the red for the weights and costs log. The added strength that made it possible to pass the test represented, in fact, an increase of 4% in curb weight and 3% in cost with respect to the reference car (500 mod.).

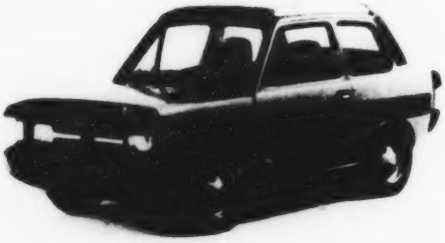


Figure 39

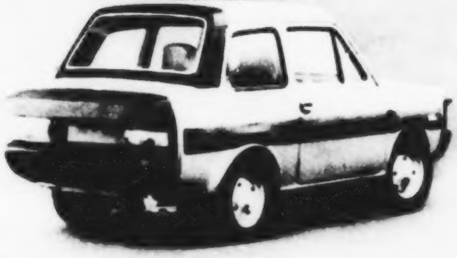


Figure 40

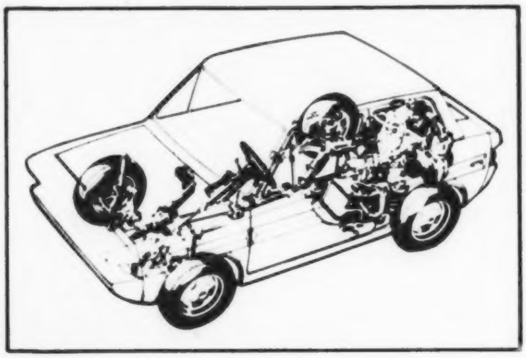


Figure 41

A second objective was the evaluation of the optimum compromise condition between the requirements of safety – in terms of survival space – and of techno-economical motivations – in reference to the particular class of car considered. In other words, the aim was the search for a severity level for collision tests, such as would ensure reasonable car protection features while avoiding an excessive penalization in

Figure 42

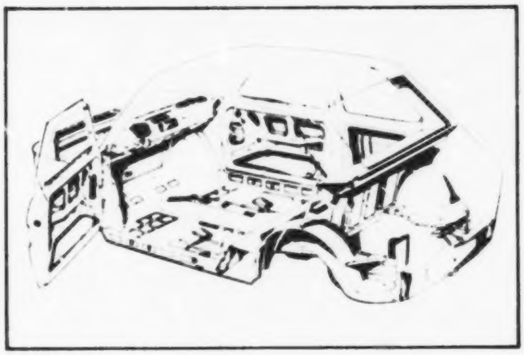


Figure 43

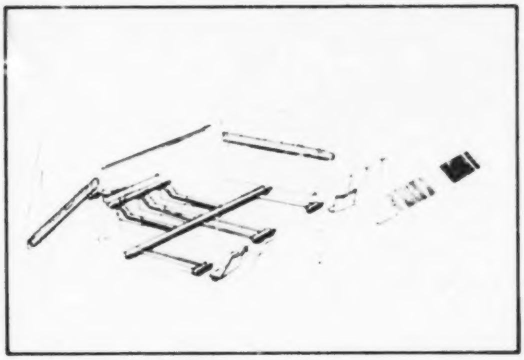


Figure 44

car weight and cost. In view of this, a set of front-and-rear collision tests at speeds below 50 mph, was programmed, with cars having structural characteristics presumably proportional to the chosen reduced severity levels.

For a first barrier front-end collision test at 40 mph, the 126 Sedan was selected (Figure 42). This Fiat model, for which mass production was started only a few months ago, differs from the 500 Sedan because of its new design body and some improvements in mechanics, one being the engine displacement increase from 500 to 600 cc. The body of this new car (Figures 43 and 44) was designed to reflect our latest safety structure experience. This has permitted remarkable improvements – with respect to the 500 Sedan (still in production) – in barrier collision performance at 30 mph, limiting the consequent body weight increase to 30 Kg only. The 40 mph test, however, showed that the improvement achieved was still not sufficient to ensure the crashworthiness required by the new severity level.

A second front-end collision test at 40 mph had as its objective, a prototype of the 1,500 pound ESV, of a special version, obtained by eliminating or lightening some structural components.

The prototype built in this third version was 175 pounds lighter than its predecessor, thus reducing to about 35% the total weight increase compared with the standard 500 Sedan. However, the test did not yield results meeting expectations in regard to the recorded values for both maximum intrusion and body deformation.

The negative result was disappointing, all the more so as it had been obtained with a lower weight gain than the one which would have been reasonable to expect from the selected severity level.

PROTOTYPES	VERSION	WEIGHT
No. 1 No. 2 No. 3	A	+ 46 %
No. 4 No. 5	B (reinforced)	+ 50 %
No. 6	C (lightened)	+ 35 %

A third aim was to improve the car protection system at low impact speeds. Following a critical analysis of the features and performance of the two initially tested systems, it was decided to proceed with the research work only on the solid foam type (Figure 45) which, compared with the pneumatic type (Figure 46), offered more interesting application prospects thanks to its simpler construction and greater freedom from shape limitations. In detail, the front and rear bumper systems for the ESV were improved by introducing the following modifications (Figure 47):

- Lightening of the foam section by a number of suitably spaced, truncated cone-shaped cavities, with open end on the bumper rest face. This was to provide the bumper system with the desired flexibility and thus overcome the practical difficulty of obtaining foams with densities lower than a given limit value and of controlling various foam densities in different areas, according to bumper styling requirements and the nonuniform crushing strength of both fore and aft terminal ends.

- Provision of ample vertical chutes in bumper lower section under the lighting system to prevent snow or dirt build-up on lamp units.
- Splitting of the bumper in two parts, upper and lower, for the dual purpose of simplifying construction and improving the geometrical visibility characteristics of lighting devices.

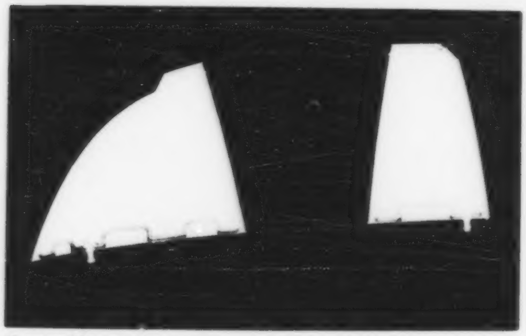


Figure 45

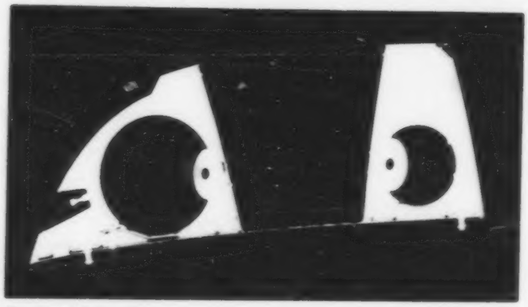


Figure 46

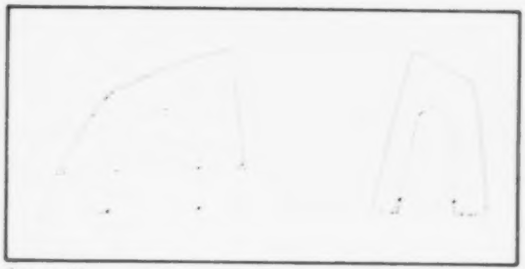


Figure 47

The beltline side protection bands were slightly modified in shape and thickness.

The roof panel rear end was modified to stress, styling-wise, the roll bar function.

The overall car dimensions (Figures 48 and 49) have not changed appreciably compared with the first prototype displayed in Washington. Referring to the basic 500 model, percentage increases in weight and cost are presently as follows:

	WEIGHT	COST
Bodywork structural systems	+ 33 %	+ 43 %
Front, rear and side bumpers	+ 9 %	
Mechanical components	+ 8 %	
TOTAL	+ 50 %	+ 43 %

Therefore, the car curb weight is 1,720 pounds.

As stated in Washington, the main scope of the 2nd phase in the 1,500 pound ESV development was the assessment of the possibility of meeting — with an appropriate restraint system — the occupant survival requirement as defined in the DOT Specification for the ESV. The results obtained in this particular field of our research and development activity are submitted in a separate report of this presentation.

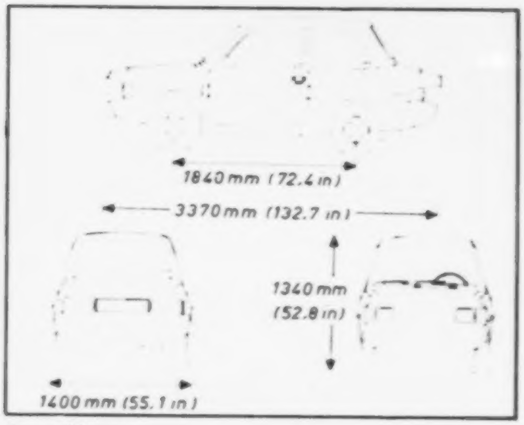


Figure 48

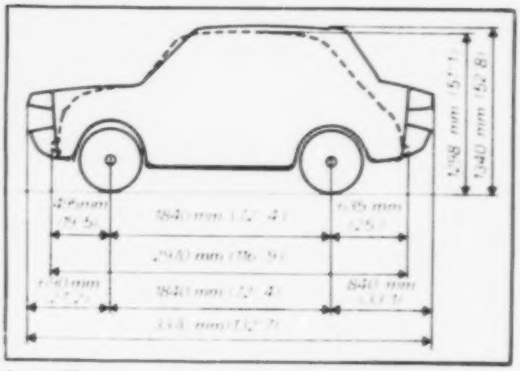


Figure 49

The 2,000 Pound ESV (Figures 51, 51a and 52)

At the Second ESV Conference in Stuttgart, Fiat presented its motor vehicle safety research program (Figure 50) and stated it would progress in three different directions. The purpose of this approach was to arrive at the definition of the specifications and features of three ESVs in the 1,500 pound 2,000 pound and 2,500 pound weight classes, derived from three typical current production cars belonging to the 1,200 pound, 1,800 pound and 2,200 pound classes respectively.

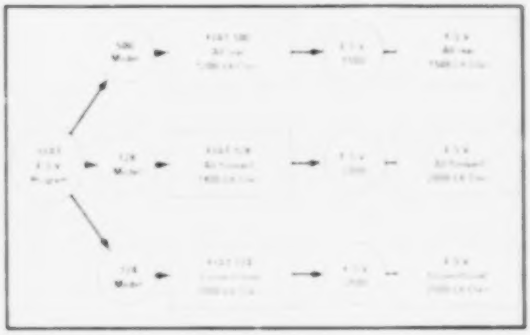


Figure 50

After having built, notably in advance of the deadline dates, the first series of 1,500 pound ESV prototypes, the time came for the 2,000 pound ESV.

The model on which the study of this new prototype was based is the 128 (Figure 53), one of the most popular cars in Fiat's present line. Briefly, it is a four-door unitized, front-wheel drive sedan with transverse power plant at front, the engine being an 1,100 cc, 55 HP unit. Top speed is about 90 mph and curb weight is just under 1,800 pounds. Its dimensional, technical and economical features are those of a typical medium-low class European car.

After having concluded the preliminary experimental cycle on numerous standard and reinforced units of this model, initial studies were commenced in June of last year to define the project fundamentals for the 2,000 pound ESV.

Contrary to what happened for the 1,500 pound ESV, this time it was decided to conduct a true study of form of the new car in order to single out possible incompatibilities between styling requirements and safety performance. At the end of July, the clashing requirements of the Design Department and the Styling Center found a meeting point in one of the many proposed models (Figure 54). Consequently, design work for the new car was given the "go" signal, early in September.

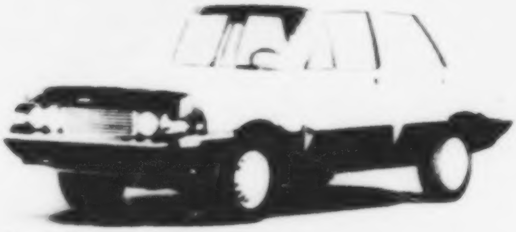


Figure 51

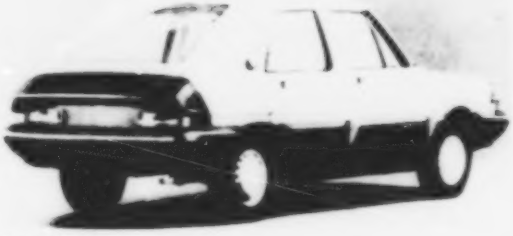


Figure 51a

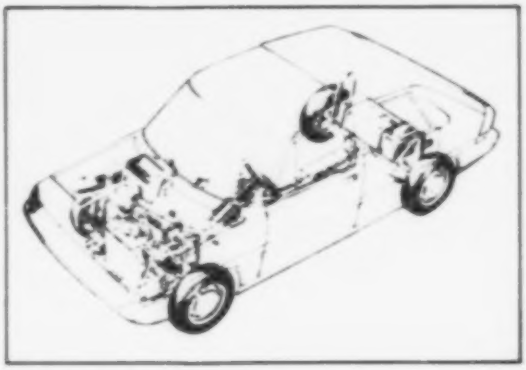


Figure 52

As the target was to exhibit the new prototype and report on test results at this conference, about six months lead time was left to complete the three project development stages (design, construction and testing). So far, five prototypes of the 2,000 pound ESV have been built, four of which were destroyed for testing and one exhibited at this conference.

The new ESV is a four-door family sedan, having still acceptable outer dimensions, even if considerably larger than the basic car (Figures 55 and 56). The overall length is 4,215 mm, width 1,690 mm and a height of 1,405 mm. A comparison with the 128 Sedan, of which the ESV retains the same 2,448 mm wheelbase, shows an increase of 10% in length, 6% in width and 4% in height.

Externally, the car body features large front and rear bumpers, as well as wide and thick underbelt

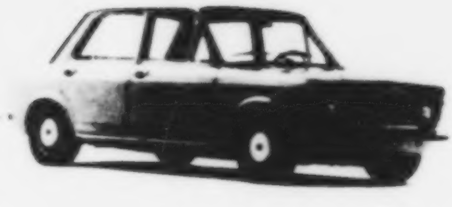


Figure 53

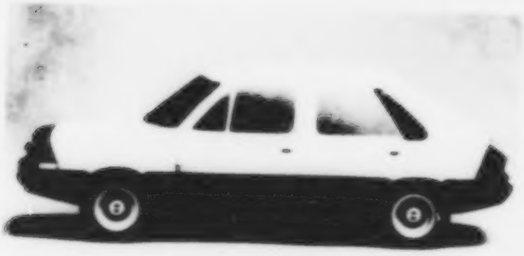


Figure 54

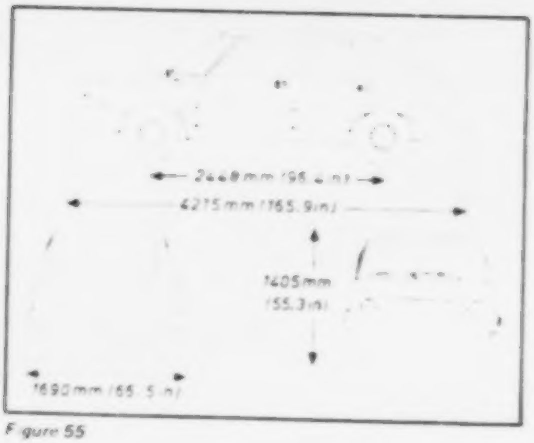


Figure 55

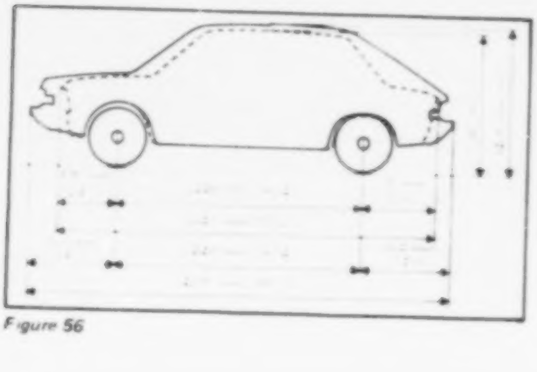


Figure 56

protection bands along the whole sides. These protection components have been made according to the same techniques and taking the same provisions as for the 1,500 pound ESV. The front and rear bumpers protrude about 200 mm from the bodyshell mounting surface and side band thickness is approximately 50 mm. By providing a deeply curved windshield and retaining the A-pillar position as in the basic car, the driver's visibility requirements have been met. This also ensures sufficient room in the upper area ahead of the front seating positions and reduces the likelihood of front occupants hitting their heads against the windshield, provided suitable restraints are adopted. The roof is slightly raised in the rear half, this styling expedient being dictated by two different requirements: first, to permit housing a sturdy roof bow connecting the two wide B pillars inside the passenger compartment and, second, to increase the back glazing height for greater visibility. The main reason for the large tailgate, not provided in the 128 basic car, is to provide an emergency egress enhancing the possibility of escape or rescue of the occupants in the event of a crash. The addition of the tailgate has also permitted the adoption of the particular rear end configuration without which it would not have been possible to obtain a sufficiently large opening between the fixed back window and the rear bumper system for access to the trunk.

As for the mechanical components, they are practically the same as for the 128. The only differences are those arising from the adoption of an engine with piston displacement increased to 1,300 cc., larger tires (155 x 13), a new fuel tank, a new radiator, suspension springs suited to the higher loads and slightly modified hand and foot controls for installation requirements. It should be noted, however, that such modifications are only a few of those which will subsequently be needed to bring the car and its single mechanical components to the previous performance and life levels.

The arrangements of the various components is practically unchanged. It is easy to understand the reasons for moving the spare wheel from the engine compartment to the trunk and the fuel tank from below the trunk to the area immediately behind the passenger compartment.

The structure of the new bodywork (Figure 57) has been suggested by the same criteria which led us to define the 1,500 pound ESV structural arrangement whose full validity was proven by the tests carried out. Also in this case, three structure systems are identifiable: one at floor level (Figure 58), a second at the belt line (Figure 59) and a third at the

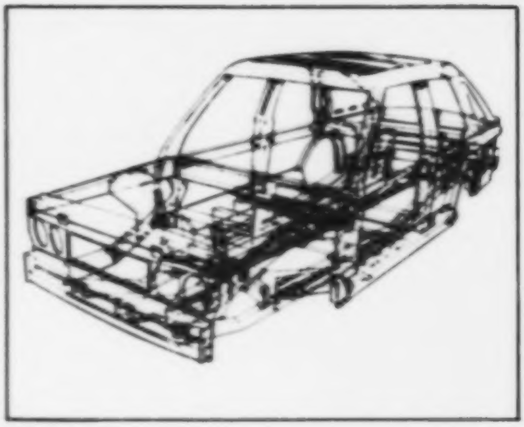


Figure 57

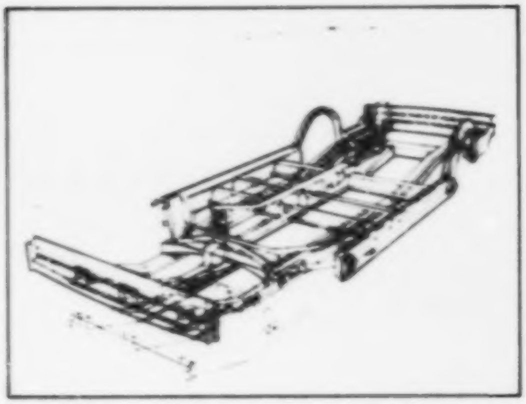
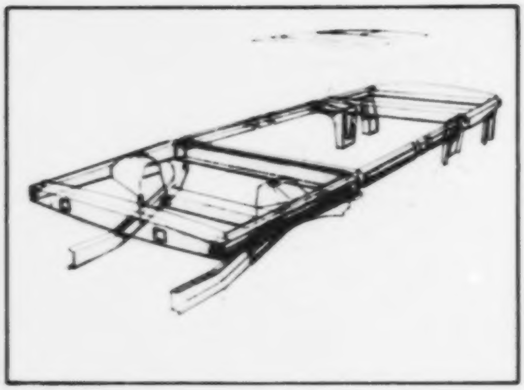


Figure 58

roof (Figure 60). However, the power plant front transverse arrangement and the ensuing serious space limitations at front, have required some major modifications, mainly in the primary (lower) structure system. In this connection it is worth observing the sturdy detachable boxed frame which supports the engine/transmission unit at center and connects the floor structure members to the front end lower crossmember.

Contrary to expectations, the results of the tests carried out with the first prototype, mainly the front-end collision test against barrier, were unsatisfactory, whereby a lot of modifications were necessary on the subsequent prototypes. Among other things, the negative consequences of a number of concessions to styling appeared evident at several points.

The results obtained in the latest tests will be stated further on. They represent the current progress stage of our research and are hence susceptible of further improvements.



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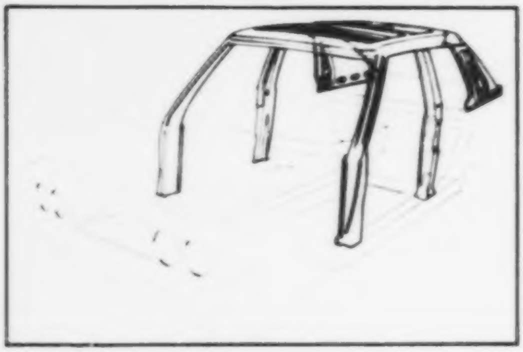


Figure 60

The interim results of the first phase of research on the 2,000 pound ESV, whose schemes — in accordance with the set work schedule — were exclusively the "survival space," "low speed protection" and "aggressiveness," are summarized below. With respect to the standard 128 model, the 2,000 pound ESV percent cost and weight increases, due account being taken of the expectable consequences on the mechanical components, are as follows:

	WEIGHT	COST
Bodywork structural systems	+ 30 %	+ 40 %
Front, rear and side bumpers	+ 6.5 %	
Mechanical components	+ 8.5 %	
TOTAL	+ 45 %	+ 40 %

The car curb weight, keeping the same accessory and trim weight as the 128 Sedan, would be of 2,570 pounds.

2,500 Pound ESV (Figures 61, 62, and 63).

During the period in which the first phase of the 2,000 pound ESV program was started and con-

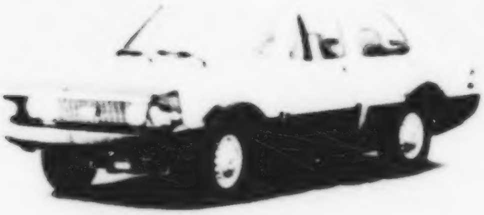


Figure 61

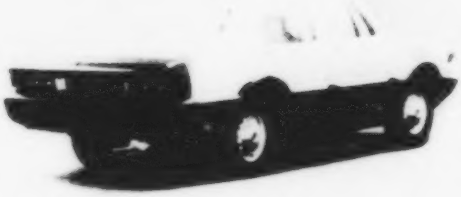


Figure 62

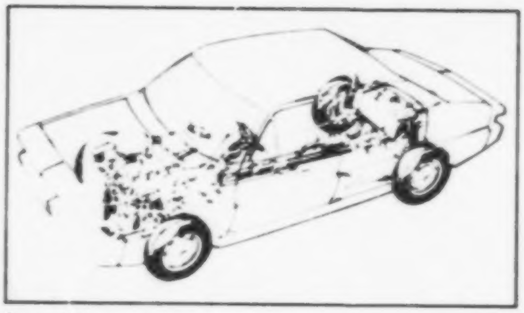


Figure 63

cluded, the same Engineering Department engaged in this work developed a similar program on the 2,500 pound ESV. The design, manufacture and test schedule was delayed, with respect to the 2,000 pound ESV, approximately 15 days as far as the start and intermediate dates were concerned, whereas the final date, set also in this case to coincide with this conference, was the same.

The car from which the 2,500 pound ESV project has been derived is the 124 model (Figure 64), a conventional design four-door sedan with front longitudinal engine and rear drive. In its basic version, the car, equipped with a 65 HP 1,200 cc. engine, has a maximum speed of about 95 mph. Curb weight is about 2,000 pounds.

According to the European classification, this car can be considered as belonging to the medium class. Its restyling for the ESV project, jointly carried out by the Styling Department and the Engineering Department (Figure 65), has transformed the vehicle into a four-door family sedan of modern line but with classical arrangement. The more peculiar components are still the front and rear bumpers and the side protection bands that present again — in new style — the solution adopted in the two previous ESV's for car bodyshell protection in low-speed impacts.

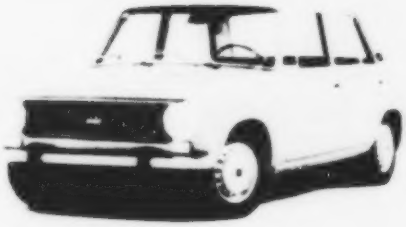


Figure 64

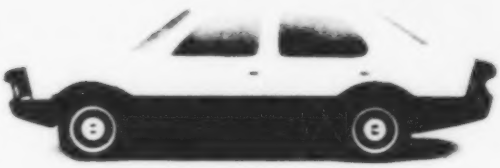


Figure 65

The deeply-curved windshield, the sturdy B pillars and the tailgate are a replica of the provisions taken in the 2,000 pound ESV, whereas the greater height of the roof is, in this case, concealed by some stylistic expedients.

The car wheelbase of 2,420 mm is the same as for the 124 Sedan; the overall length is 4,375 mm, width 1,730 mm and height 1,430 mm (Figure 66).

In comparison to the 124 Sedan, this ESV is 8% longer, 6% wider and 5% higher (Figure 67).

The modifications incorporated in the basic 124 model mechanical components are the same as those listed for the car derived from the 128 model: larger tires (175 x 13), increased piston displacement to 1,450 cc., new radiator, new suspension springs,

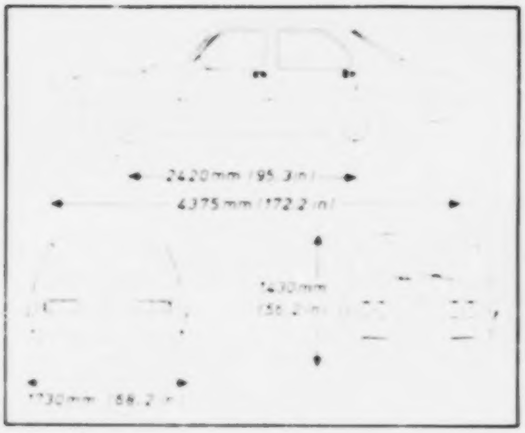


Figure 66

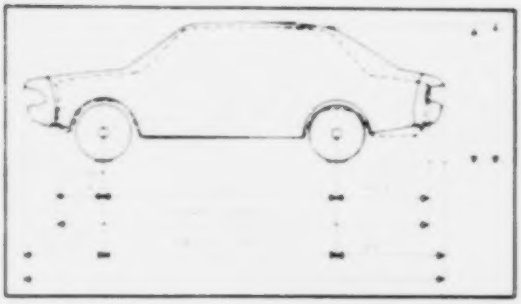


Figure 67

modifications of hand and foot controls, new fuel tank displaced to the area immediately behind the passenger compartment and rearrangement of the spare wheel at the bottom of the trunk.

The bodywork structure (Figures 68, 69, 70, 71) follows the line adopted for the 2,000 pound car with some variants due mainly to the longitudinal arrangement of the power plant and to the propeller shaft connecting this unit to the rear axle.

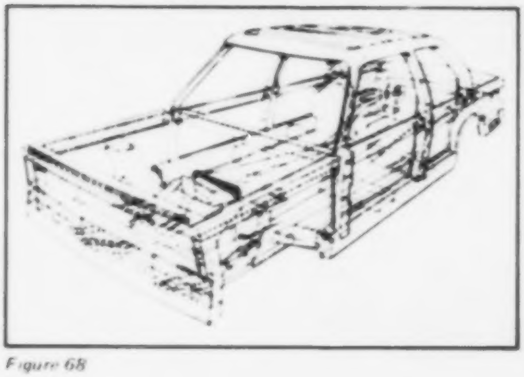


Figure 68

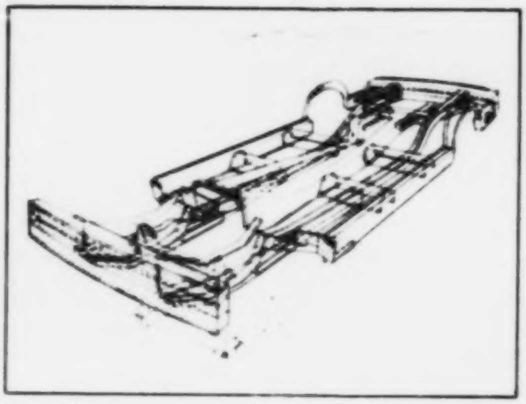


Figure 69

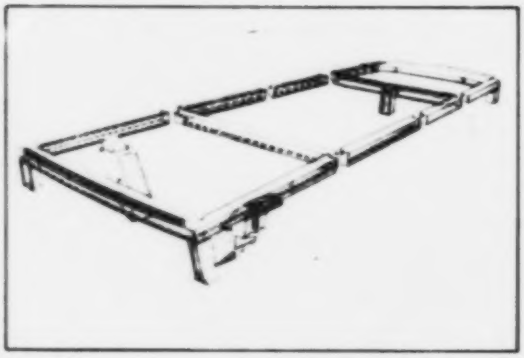


Figure 70

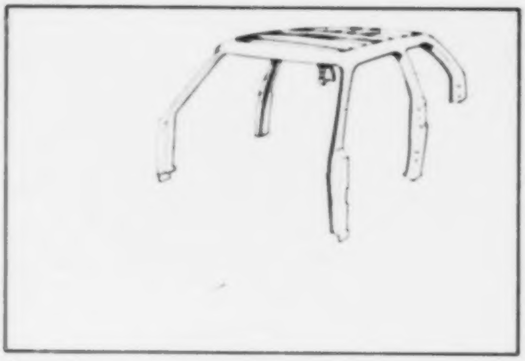


Figure 71

The lower primary structure includes, in the front section, a detachable subframe consisting of two side members and a cross-member supporting the front suspension components.

Also with this car, the same difficulties experienced with the 2,000 pound ESV were encountered in the course of the tests. In fact, the front-end collision test of the first prototype gave unsatisfactory results, mainly from a qualitative standpoint. Furthermore, the first rear end collision test run on the same prototype gave negative results. As with the 2,000 pound ESV, several modifications in the structure critical areas proved necessary; and these modifications were gradually introduced in the two subsequent collision-tested prototypes.

The 2,500 pound ESV vs. 124 Sedan weight and cost increases are outlined below:

	WEIGHT	COST
Bodywork structural systems	+ 27 %	+ 37 %
Front, rear and side bumpers	+ 7.5 %	
Mechanical components	+ 8.5 %	
TOTAL	+ 43 %	+ 37 %

The vehicle curb weight, based on the same hypothesis made for the 2,000 pound ESV, is 2,840 pounds.

As in the case of the 2,500 and 2,000 pound ESV the weight and cost increases are as could be expected more contained than for the 1,500 pound ESV.

	WEIGHT	COST
ESV 1500	+ 50 %	+ 43 %
ESV 2000	+ 45 %	+ 40 %
ESV 2500	+ 43 %	+ 37 %

In our opinion, however, they are not such as to warrant any satisfaction or optimism. Conversely, it appears that the above figures confirm our initial worries and reduce the hopes of attaining, in the future, more encouraging results than attained so far.

FIAT ESV TESTS

1,500 Pound ESV

At the Third Conference we presented the results of several tests, with the exception of against a pole at 50 mph. Owing to the particular severity of this test, local reinforcements were added on the car. Maximum intrusion into passenger compartment was 5 cm at footboard (Figure 72).

The aim, obviously, was the possibility of a compromise between maintaining the survival space and an acceptable cost. To this end, a second set of front-and rear-end collision tests at a reduced speed of 40 mph was conducted.

The standard production 126 car was tested for comparison; passenger compartment intrusion values were unacceptable.

Tests on the lightened 1,500 pound ESV have shown that maximum passenger compartment intrusion values are higher than recorded on the original ESV at 50 mph and exceed U. S. Specification values (Figure 73).

A similar situation was assessed in the rear-end collision test (Figure 74), which shows why the set goal could not be reached.

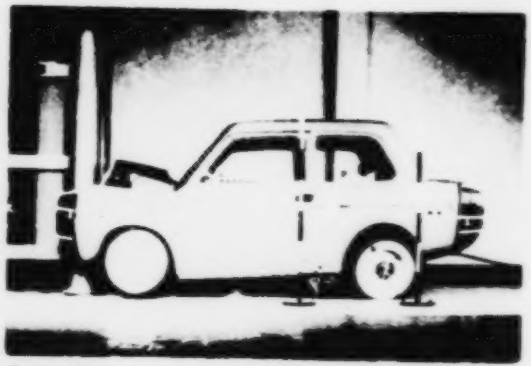


Figure 72

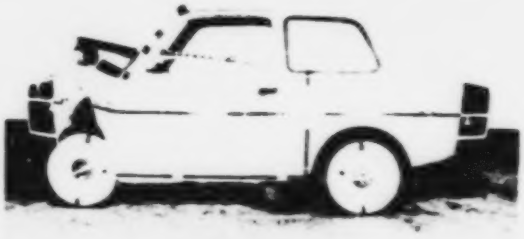


Figure 73

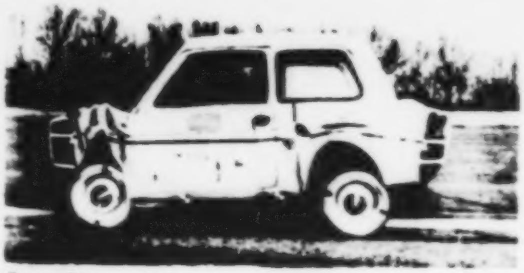


Figure 74

2,000 Pound ESV

Tests were run on the *first unit* to check the bumper and its support structure performance at low and then progressively higher speeds. They showed that the car can withstand front and rear impacts up to 10 mph without any harm.

The barrier front-end collision at 50 mph, though, gave unsatisfactory results: intrusion was 13 cm at footboard and 7 cm at instrument panel (Figure 75). Maximum deceleration during impact was 58 g as shown in Figure 76.

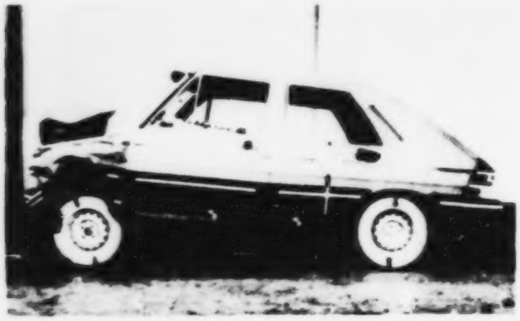


Figure 75

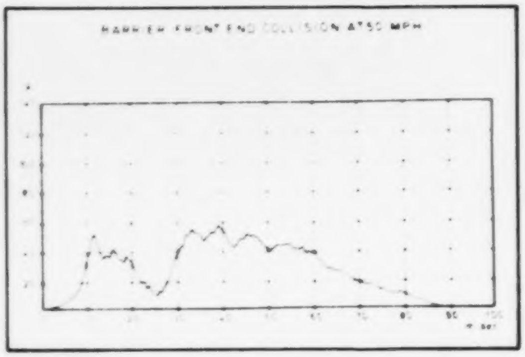


Figure 76

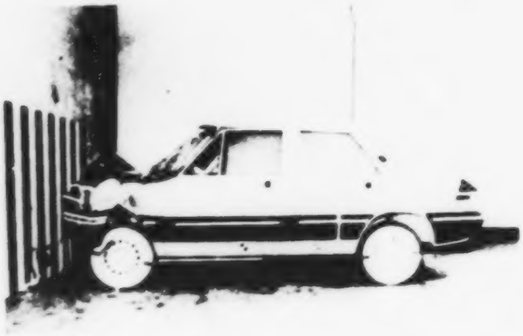


Figure 77

A highly reinforced *second unit* was then built (Figure 77). With this, passenger compartment intrusion was practically nil at footboard and just 1 cm at instrument panel, but maximum deceleration went up to 122 g (Figure 78).

The rear-end collision at 50 mph was run on the same unit. Maximum intrusion remained limited to 3 cm (Figure 79).

The side impact test at 15 mph was run on the same unit. A maximum intrusion of 16 cm was measured at wheelbase mid-point; it dropped to 10 cm on front door longitudinal center, and to 6 cm on the B pillar (Figure 80).

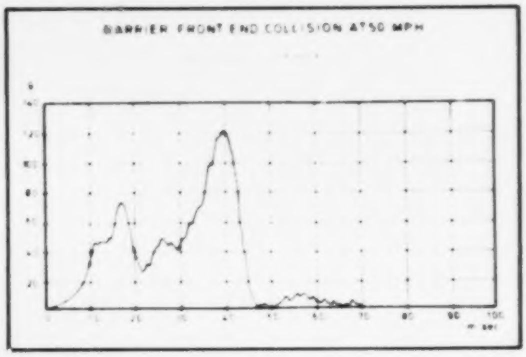


Figure 78

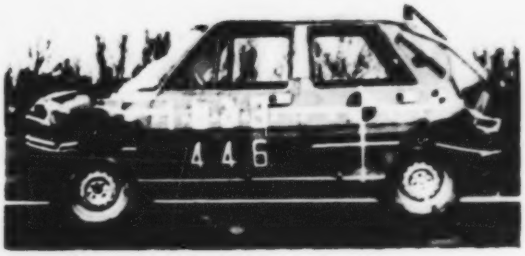


Figure 79

A compromise was sought on a *third unit* between passenger compartment intrusion and deceleration level, through a lightening of the structures of the second unit.

In the 50 mph barrier front end collision test, intrusion was 2.5 cm at instrument panel and 1 cm at footboard, while deceleration was reduced to 75 g (Figures 81 and 82).

The simulated rollover test was carried out on the same unit by means of a fall from a height of 60 cm. Maximum intrusion was 9 cm (Figure 83).

A *fourth unit* was set up to conclude the test cycle by permitting a greater intrusion than on the third

Figure 80

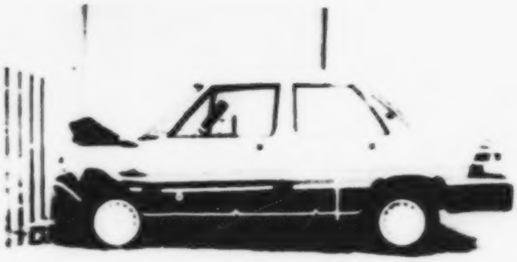


Figure 81

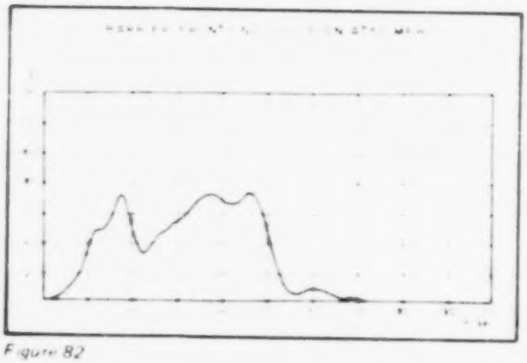


Figure 82

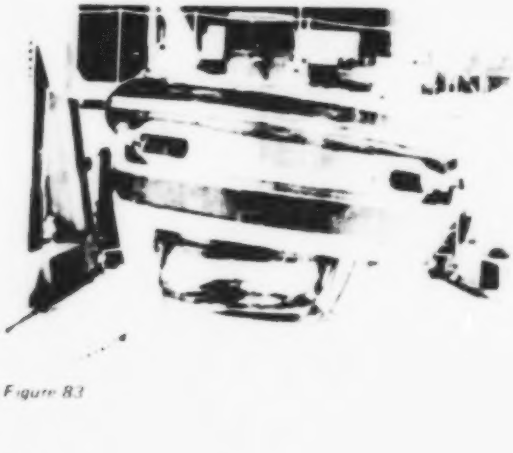


Figure 83

unit, but reducing maximum deceleration (Figure 84).

The results sought were attained, as follows: 3 cm intrusion at instrument panel and 1.5 cm intrusion at footboard, with a maximum deceleration of 70 g (Figure 85). It is worth comparing the decelerations obtained on the different units as a function of the maximum intrusion recorded at instrument panel and footboard.

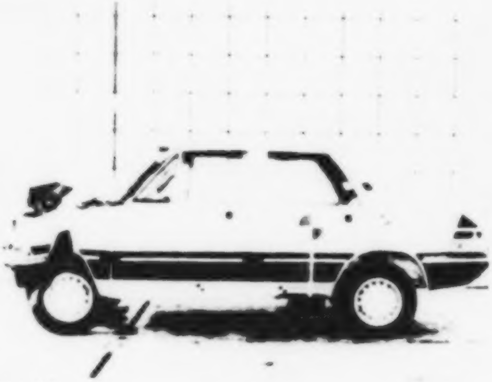


Figure 84

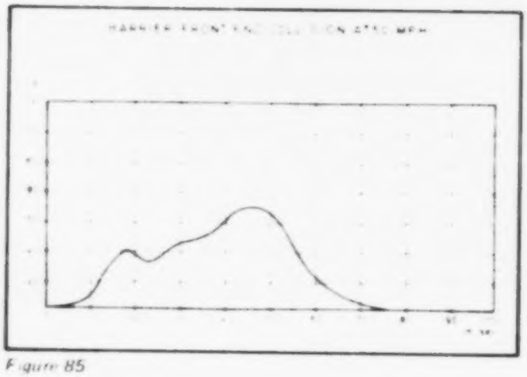


Figure 85

The chart of Figure 86 shows that if intrusion is to be contained as the Specification requires, it is not possible to go below deceleration values of 70 g for this car.

2,500 Pound ESV

A front end collision test at 50 mph was run on the first unit (Figure 87). Maximum intrusions recorded at instrument panel and at footboard were 7 cm and 10 cm respectively. Maximum deceleration was 76 g (Figure 88).

The structure was reinforced on the *second unit*, no intrusion occurred at instrument panel in the front

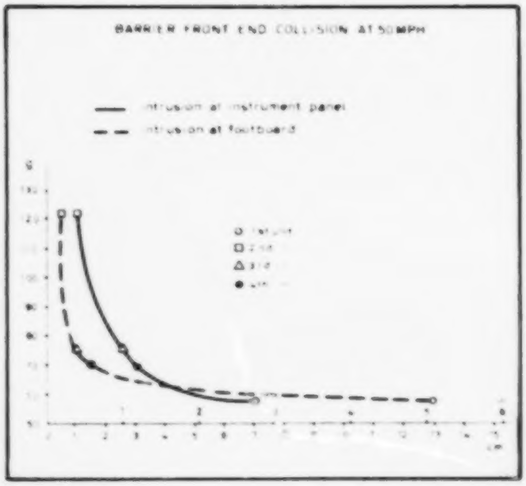


Figure 86

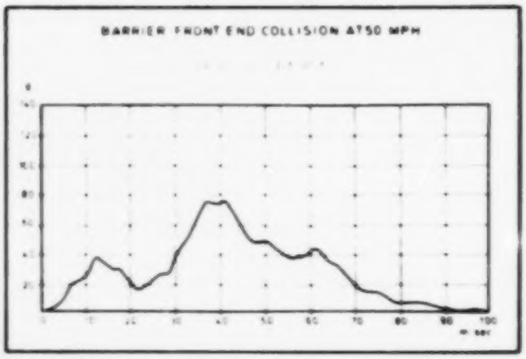


Figure 88

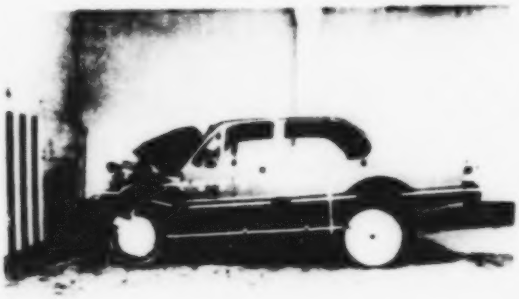


Figure 87

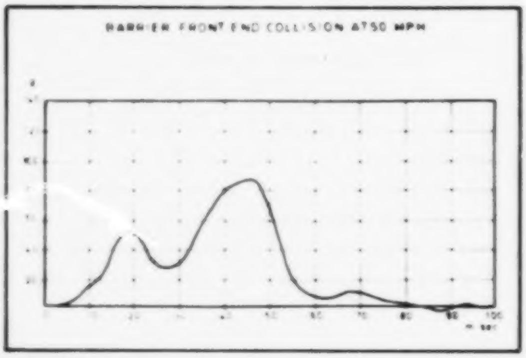


Figure 89

end collision while at footboard it was contained at 8 cm (Figure 89). Deceleration, instead, went up to 89 g (Figure 90).

The rear-end collision test at 50 mph was carried out on this same unit; maximum intrusion was 12 cm, a still excessive value (Figure 91). We think that with minor modifications the required limits can be met.

The *third unit*, with the same modifications as for the second, was submitted to the side-impact test against pole at 15 mph; maximum intrusion resulted 13.7 cm at wheelbase midpoint; it dropped to 10.5 cm at front door longitudinal center and 8.3 cm at B pillar (Figure 92).

We think that in this test, too, we can meet, with slight modifications, the maximum intrusion requirements.

Finally, the simulated rollover 60 cm fall test was run on the same unit; maximum intrusion was 3.7 cm, which is highly satisfactory (Figure 93).

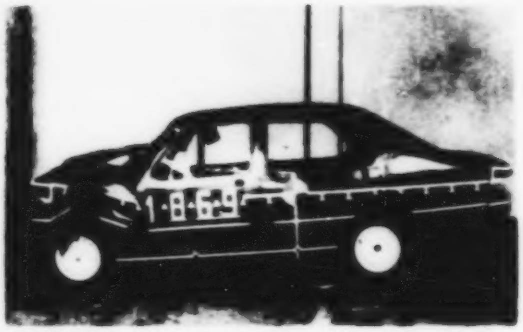


Figure 90

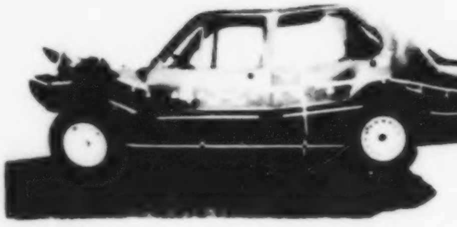


Figure 91

Figure 92

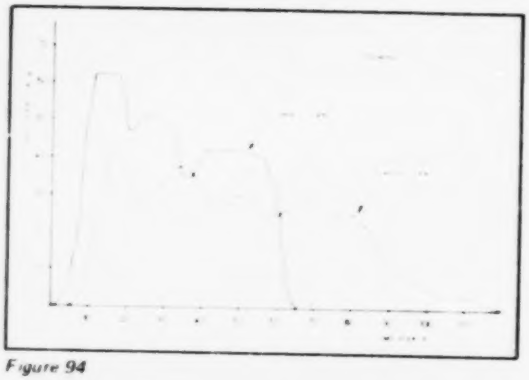


Figure 94

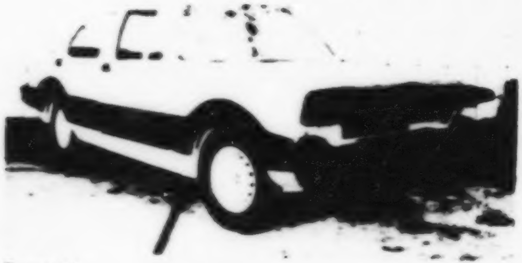


Figure 93

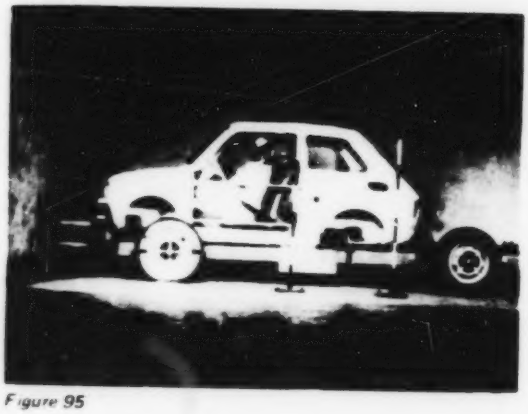


Figure 95

RESEARCH ON RESTRAINT SYSTEMS

After the survival space tests conducted on the 1,500 pound ESV, we tackled the problem of restraint systems for it by trying to solve the problem using the deceleration values measured with our ESV's. In fact, it seemed illogical to work on levels of 40 g because this value is unattainable on a small car unless its size is increased drastically. As a matter of fact, in our ESV, as built, two serious difficulties were encountered:

- Small passenger compartment
- High deceleration level (Figure 94).

The problem was therefore more complex than for larger sized cars.

Tests With Seat Belts

Tests were run on a bogey-mounted 126 bodywork at increasing speeds. The bogey was launched by a catapult and decelerated by the deformation of front tubing (Figure 95). The deceleration levels and patterns found in the barrier tests were thus reproduced.

Conventional Belts Without Retractor

The following tests were run with conventional 3-point belts, without retractor and fitted snug on a 50th percentile dummy. The results are shown in the Table of Figure 96 along with the results of the other tests.

- Test 1735/C: at 30 mph, even without hitting the passenger compartment interior, dummy deceleration exceeded the permissible values.
- Test 1753/C: when the speed was raised to 40 mph, the dummy hit the structure and in some cases belt failure occurred; obviously, upon dummy impact very high deceleration values were recorded.
- Test 1749/C: on repeating the 40 mph test, but with a belt webbing having an inextensibility value three times as great, dummy impact could not be prevented.

Conventional Belts With Retractor

The following tests were run to analyze the influence of retractors:

Test No.	Restraint System	Test Device	Speed	Dummy decelerations						Femur Load	
				Head		Thorax		Pelvis		Left	Right
				Hor.	Vert.	Hor.	Vert.	Hor.	Vert.		
			mph	g		g		g		lbs	
[illegible]	Belts,[illegible] elongation	Bogey,124 body	30	76	64	58	30	44	82	445	270
[illegible]	Belts,[illegible] elongation	Bogey,124 body	40	200	230	82	92	72	112	600	530
[illegible]	Belts, [illegible] elongation	Bogey,124 body	40	78	172	90	35	76	90	1300	750
1736/C	Belts,[illegible] elong w/retractor	Bogey,124 body	30	64	90	54	32	59	-	1450	1200
1737/C	Belts, [illegible] elong w/retractor	Bogey,124 body	30	100	116	58	34	63	-	550	530
1769/C	Metal belts,hydr.dissipator	Bogey,124 body	30	45	60	42	34	44	43	560	-
1770/C	Metal belts,hydr.dissipator	Bogey,124 body	40	70	70	52	30	50	60	670	-
[illegible]	Metal belts,hydr.dissipator	Bogey,124 body	[illegible]	105	120	110	40	100	[illegible]	-	-
[illegible]	Metal belts,Plast.deform.dissipator	Bogey,seat	30	55	49	40	20	-	-	-	-
[illegible]	Metal belts,Plast.deform. dissipator	Bogey,seat	40	75	[illegible]	44	30	-	-	-	-
[illegible]	Metal belts,Plast.deform.dissipator	Bogey,seat	50	[illegible]	[illegible]	[illegible]	[illegible]	-	-	-	-
[illegible]	Ackles Britax support-Driver	124 car	30	83	57	37	12	-	-	-	-
	Ackles Britax support-Passenger	124 car	30	72	50	41	-	-	-	-	-
[illegible]	Ackles Britax support-Driver	Reinf. 124 car	40	70	80	60	12	-	-	-	-
	Ackles Britax support-Passenger	Reinf. 124 car	40	110	99	77	52	-	-	-	-
	Air Bag Driver	[illegible]	30	[illegible]	190	142	52	-	-	-	1000
	Air Bag Passenger	[illegible]	30	[illegible]	[illegible]	172	90	-	-	2260	1640

Figure 96

- Test 1736/C at 30 mph with standard inextensibility value belts the dummy knees hit the passenger compartment because part of the webbing came out of the retractor before it locked. Decelerations were not increased greatly but femur loads were higher.
- Test 1737/C at 30 mph, with higher inextensibility value webbing, dummy impact was prevented but decelerations increased considerably.

Our tests therefore show that for this car, conventional belts (with or without retractors) are not acceptable even at speeds of 30 mph as, to prevent dummy impact, decelerations must be raised beyond the permissible levels.

Metal Belts With Fluid-Type Energy Dissipators

The limitations encountered are essentially due to belt extensibility. We have, therefore, devised a system with high inextensibility webbing (in practice a metal strip) whose ends are connected to an energy-dissipating device which permits a controlled movement. We have used various types of devices starting from two or more small fluid-type dissipators (Figure 97).

- Test 1769/C: first test at 30 mph was carried out with only two dissipators fitted to the outboard anchor points. Dummy behavior was very good. Decelerations were considerably lower than those recorded with the systems examined previously.
- Test 1770/C: when the speed was increased to 40 mph, the dummy used up the whole space available and began to contact the steering wheel. The deceleration values were still satisfactory.
- Test 1811/C: in the subsequent 50 mph test, with three energy dissipators, stress was such as to cause the fracture of the dummy neck. The corresponding decelerations were high.

Figure 97

Metal Belts With Plastic-Deformation Energy Dissipators

We have since set up a research program with energy dissipators designed to obtain suitable stroke, load and load pattern versus speed. In the tests described below the energy dissipating device consisted of an ogive-shaped element which deformed a plastic tube (Figure 98).

For the sake of simplicity, the body shell was eliminated, the seats and belt anchorages were fixed directly to the bogey (Figure 99).

Figure 98

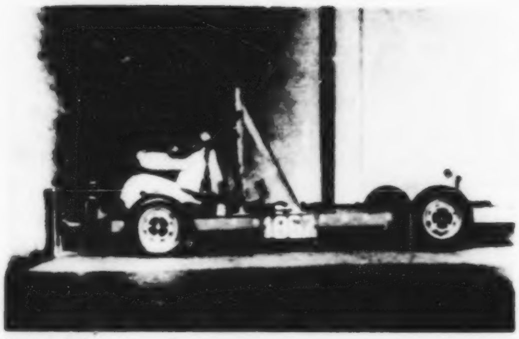


Figure 99

- Test 1854/C: in the 30 mph test with three forced penetration dissipators, an excessive pelvis travel was recorded. Decelerations were at a satisfactory level.
- Test 1862/C: in the 40 mph test, pelvis travel was reduced by relocating the lower belt anchor points. Decelerations were just a little above the permissible limits.
- Test 1867/C: no satisfactory results were yielded by the 50 mph tests, as of now. In fact, decelerations were high.

Tests With Accles Britax Supporting Devices

We carried out some tests with the Accles Britax supporting device applied on our cars (Figure 100). The system consists of a rest which acts on the occupant's chest and is connected to a vertical arm whose rotation is controlled by an energy dissipating device. This element operates only in the event of an impact.

- Test 1866/C: in the 30 mph test on a standard production 126 car (Figure 101), the system could not prevent the driver from impacting the steering wheel, conversely, passenger protection was satisfactory as the deceleration limit was not exceeded.
- Test 1864/C: in the 40 mph test run with a reinforced 500 model car (Figure 102), the driver again struck the steering wheel while the passenger was restrained; decelerations exceeded the permissible limits.

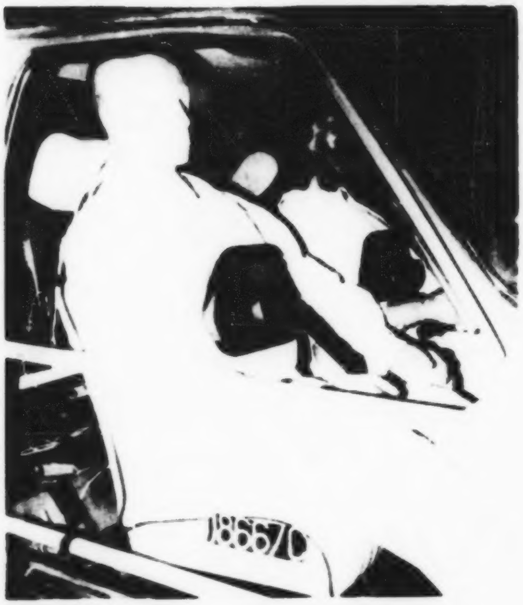


Figure 100

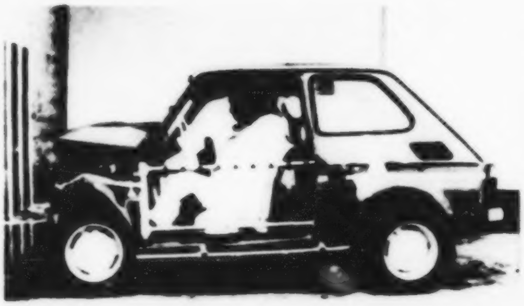


Figure 101

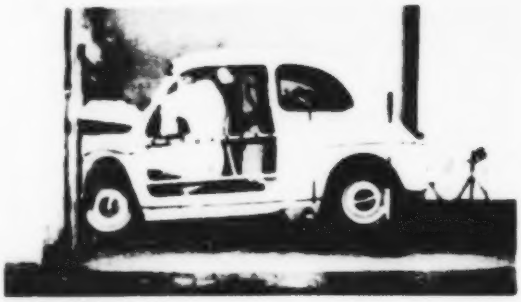


Figure 102

Tests With Air Bags

The use of air bags on the 1,500 pound ESV was also investigated. A 50 mph test was performed (Figure 103). Decelerations greatly exceeded the permissible values.

The various tests run so far lead us to conclude that for this car there is, as of now, no satisfactory restraint system for speeds of 50 mph for this ESV. The situation as regards the various systems is as follows (Figure 104).

- Conventional belts: not acceptable even at 30 mph.
- Belts with energy dissipators: their use appears to be promising and we are optimizing them for our requirements.
- Supporting devices: the possibilities of the system need further investigation.
- Air bags: their use on small cars presents serious difficulties and doubts still exist about their practical application.

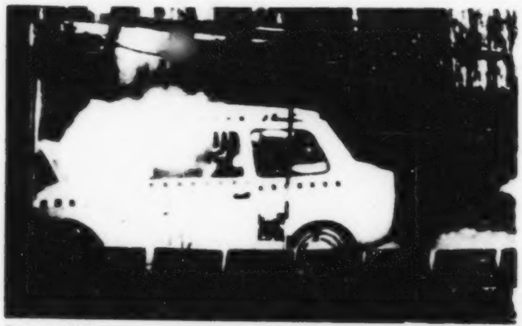


Figure 103

COMPATIBILITY

Compatibility is an impact problem and we know that the severity of an impact test is typified by two factors: velocity and type of obstacle. We have therefore studied the influence of these factors by running front-end collision tests, first against barrier and then against other vehicles.

Four types of cars of increasingly larger size have been tested, weights being 1,200, 1,800, 2,200 and 3,400 pounds. All were substantially reinforced to pass the 50 mph barrier test in terms of survival space (as we showed at the Second Conference in Stuttgart) except the 3,400 pound car which was a standard production unit.

Tests against barrier were run at 50 mph. (page 470) (Figure 105), while the car-to-car collisions were run at 44 mph (Figure 106), due to towing difficulties.

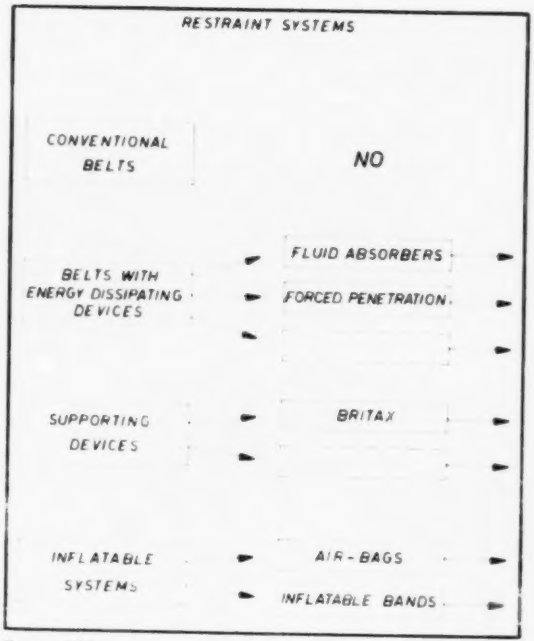


Figure 104

The pictures in Figure 106 illustrate the following car-to-car collision tests:

1,200 pound car	against the 1,800 pound car (Figure 106A)
	against the 2,200 pound car (Figure 106B)
	against the 3,400 pound car (Figure 106C)
1,800 pound car	against the 2,200 pound car (Figure 106D)
	against the 3,400 pound car (Figure 106E)
2,200 pound car	against the 3,400 pound car (Figure 106F)

Let us now analyze the overall results from this set of tests (barrier and car-to-car).

The smaller 1,200 pound car (histogram of Figure 107) always suffered more extensive damages in collisions against other types of car than against the barrier: the bigger the car it struck the greater the damage.

For the 1,800 pound medium/low class car (Figure 108), damages were less severe against the small car than in barrier impacts, but they were more severe against the larger cars. In fact, the barrier collision reproduced the conditions of a head-on impact against another car of the same type as the striking car traveling at the same velocity.

This finding is also confirmed by the histogram for the medium class 2,200 pound car (Figure 109).

The large 3,400 pound car (Figure 110) always collided against smaller ones; in fact, its damages were always less than in the barrier test and it caused the largest amount of damages to all the others.

It was then decided to try out the behavior of the 3,400 pound car against a heavier vehicle; the smallest truck built by Fiat, weighing 5,000 pounds, was used for this test. Both vehicles were speeded up to 40 mph (Figure 111A).

Car damages substantially involved the front section whereas they were not so serious inside the passenger compartment. The truck cab showed a considerable deformation on the front end panel though retaining satisfactory survival space in the interior.

As a further check, a collision was staged between a large 4,300 pound car and an unladen 14,000 pound truck; each of the two vehicles were travelling at 48 mph (Figure 111B). The large car was practically a write-off. Truck damages, on the other hand,

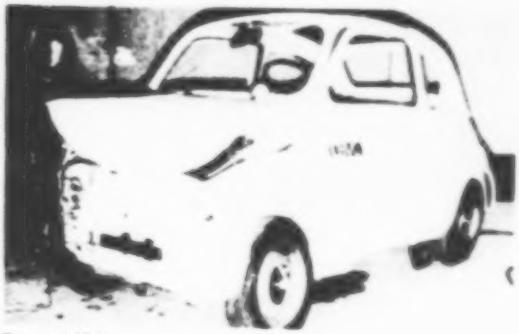


Figure 105A

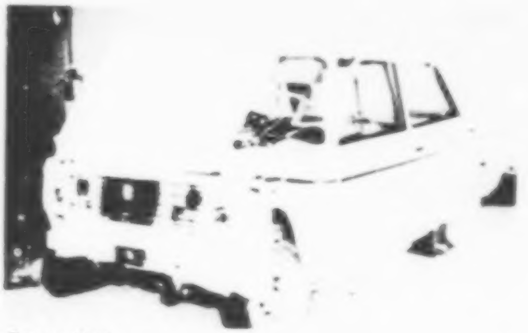


Figure 105B

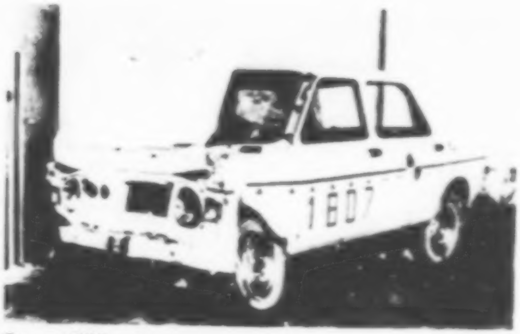


Figure 105C

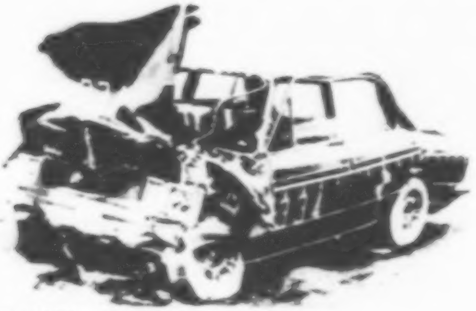


Figure 105D

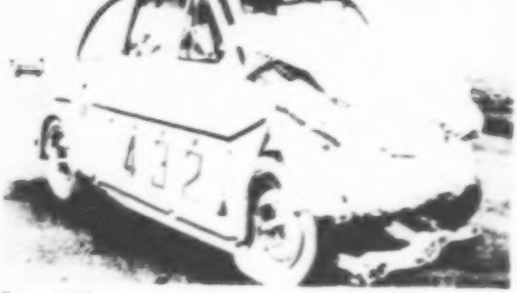


Figure 106A

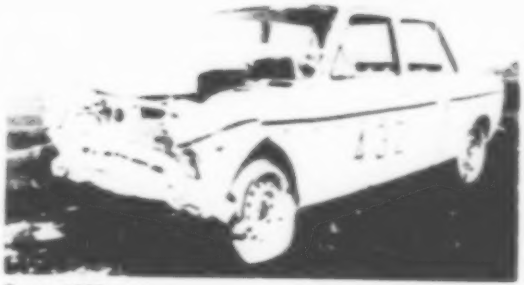


Figure 106B

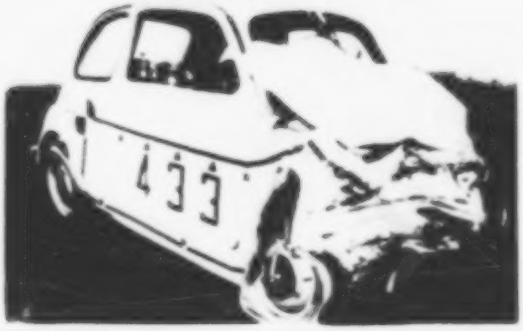


Figure 106C

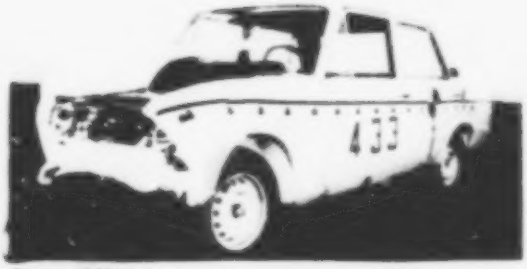


Figure 106D

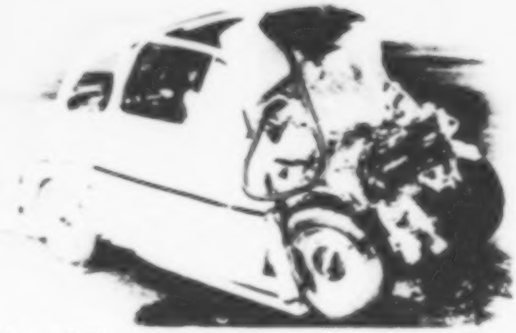


Figure 106E

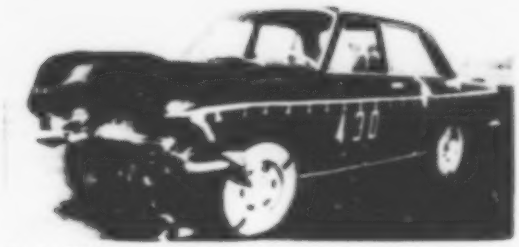


Figure 106F

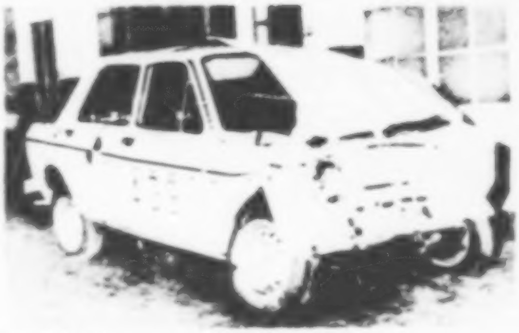


Figure 106G

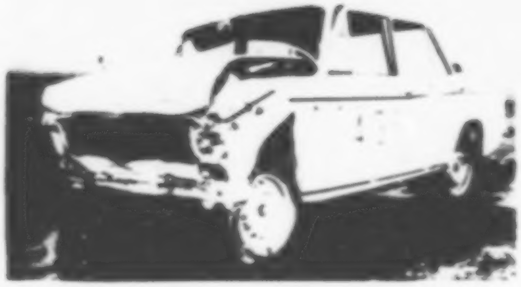


Figure 106H

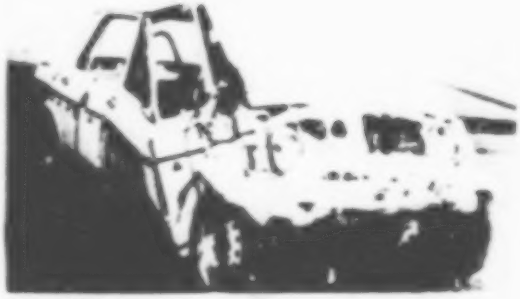


Figure 106I

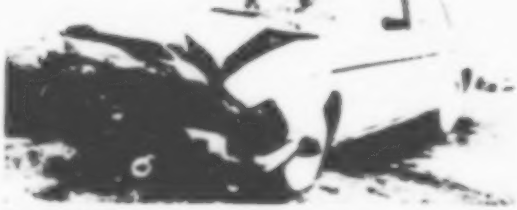


Figure 106J

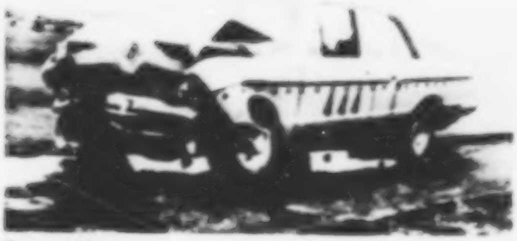


Figure 106K

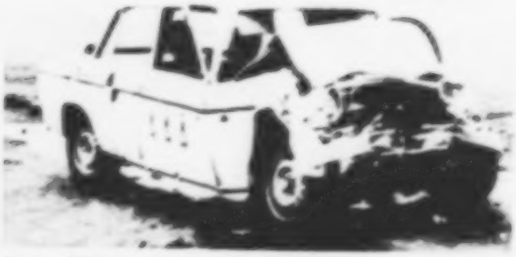


Figure 106L

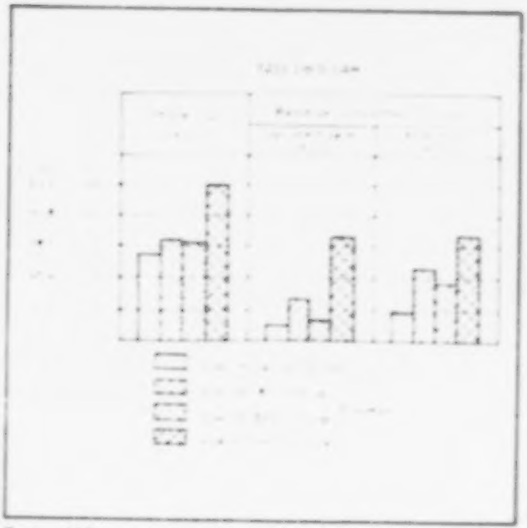


Figure 107

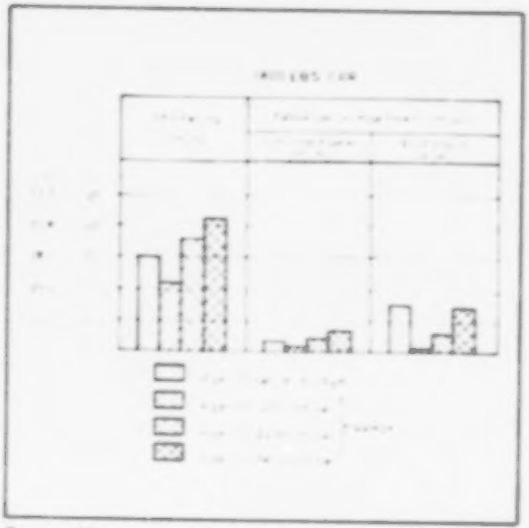


Figure 108

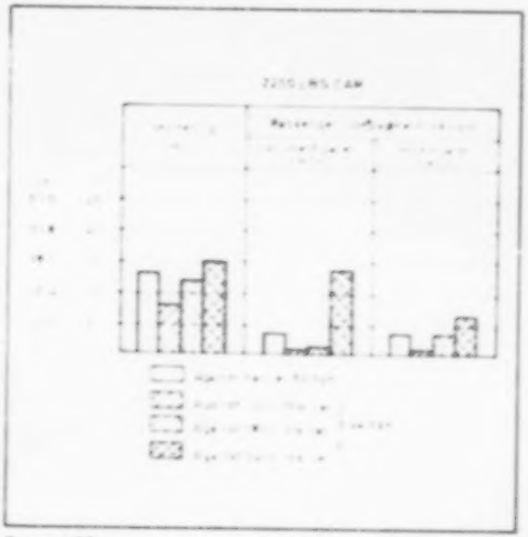


Figure 109

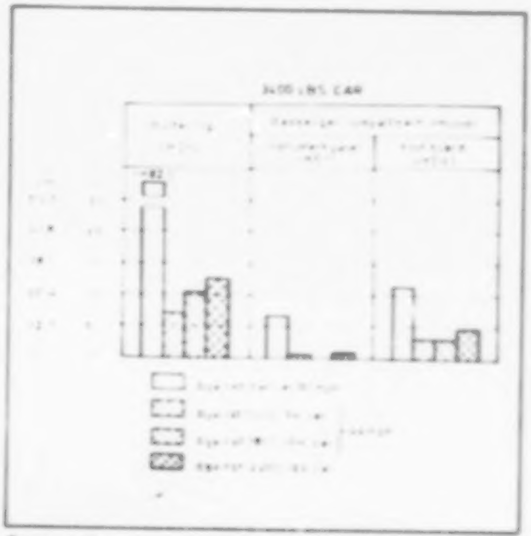


Figure 110

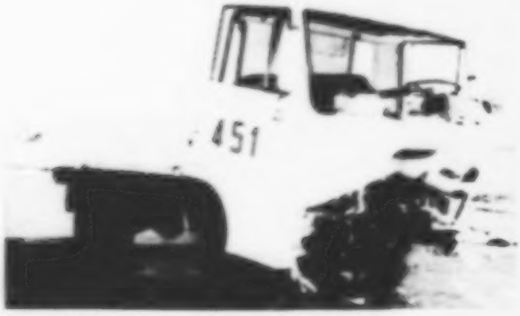
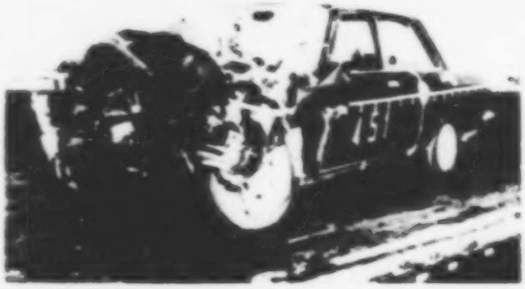


Figure 111A



Reinforced Car

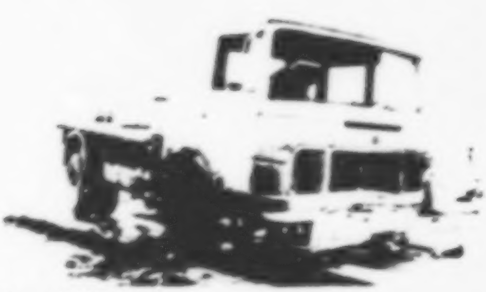
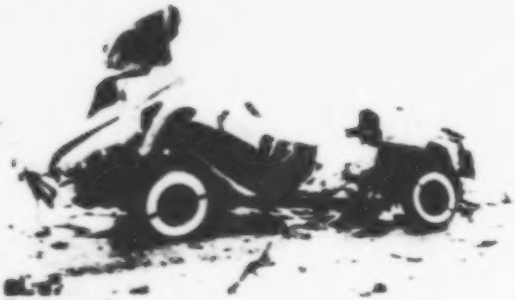


Figure 111B



Reinforced Car

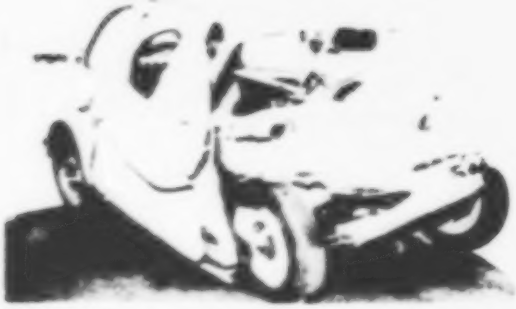
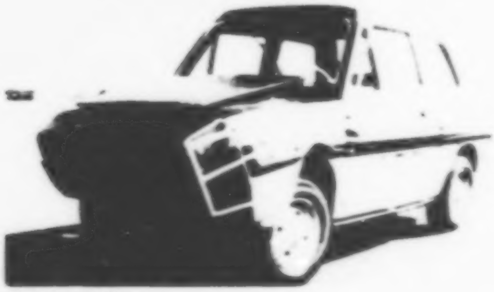


Figure 111C



Reinforced Car

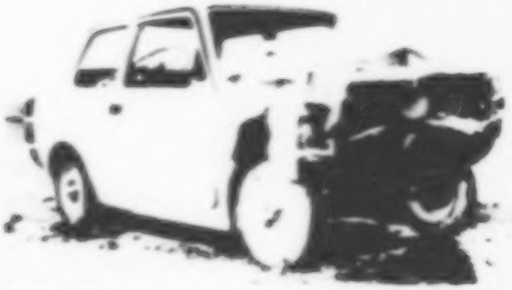
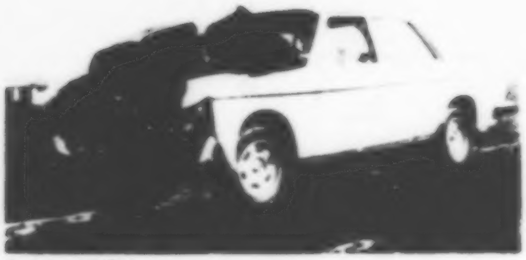


Figure 111D



Reinforced Car

involved the front suspension only. The cab was practically undamaged.

To acquire further data, the 1,500 pound ESV was launched against another car of the same weight class which had passed the 50 mph barrier test (Figure 111C). On impact, both vehicles were travelling at 44 mph. The reinforced 1,200 pound car damages were much more serious than in the test against barrier at 50 mph. The ESV damages were not only less severe than for the 1,200 pound car it struck but also less serious than those resulting from the 50 mph barrier test.

Now, let's see how it behaved in an impact with a much heavier car, the 3,400 pound unit. The test was run with both cars speeding at 44 mph (Figure 111D). The ESV car penetrated into the front structure of the bigger car, which suffered a localized deformation in the middle area of its front end though retaining a good degree of passenger compartment integrity. The ESV damages were acceptably limited and their extent may be rated as intermediate between the damages recorded in the impact with the 1,200 pound reinforced car and those against barrier.

We introduced this investigation by saying that two factors typify impact severity — velocity and type of obstacle. The tests we have described show that the obstacle itself is characterized by three factors — mass, rigidity and what we may call "interface." By interface we mean the mutual influence between car and obstacle as regards interacting shapes and the reciprocal, local distribution of rigidities (Figures 112 and 113).

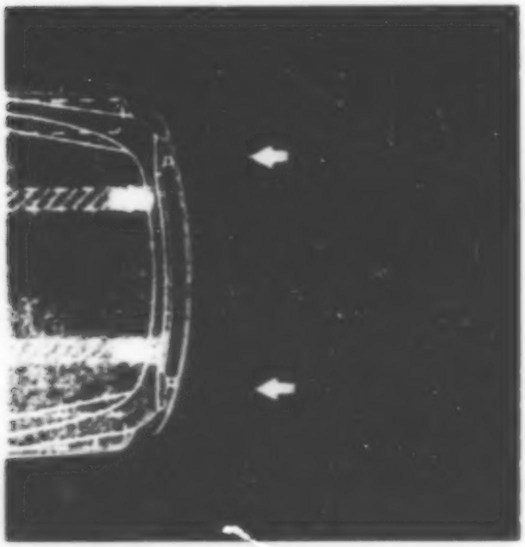


Figure 112

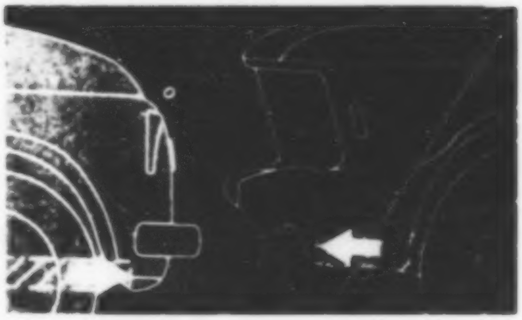


Figure 113

Compatibility is clearly a complex problem, which requires a tremendous amount of experimentation, and the widest possible collaboration.

CONCLUSIONS

The development of the first ESV prototypes in the three weight classes selected enabled us to evaluate, as realistically as possible, the consequences of compliance with the requirements of the ESV specification. These consequences are, as we have seen, rather serious, especially for smaller economic cars of the European type. However, our tests have shown that the problems of the survival space and of low speed protection are technically solvable, but at the expense of still acceptable size increases and of cost increases which in today's terms, and in the best of hypotheses, are not likely to be much less than 40%.

However, meeting the additional requirements of maximum car body deceleration might change the picture drastically and cast a shadow over the entire situation. In fact, further increases in size, weight and cost would be unavoidable and, if the Specification were enforced as a standard, one consequence would be the disappearance of the minimum weight class cars, whose social importance, particularly in Europe and in Italy, we have stressed on more than one occasion.

It is too early to say firmly, either way, whether the hope that an adequate solution can be found to the serious problem of "occupant survival" is justified by test results, at least with regard to minimum and medium weight class cars and barrier impact severity levels above 30 mph. The possibilities of significant results in this field are also considerably handicapped by the lack of effective means for evaluating the various systems under study — namely, a dummy with mechanical characteristics sufficiently close to those of the human body and capable of ensuring repeatibility of test results.

The importance of the problem of compatibility between vehicles having different mass and structural characteristics has already been shown. It appears increasingly complex and difficult to solve. However, a solution should not be altogether impossible, even if it will require further exhaustive research and will only be the result of a careful compromise for the coexistence of the various vehicles on the same roads.

We hope that the future ESV Specification promised by the DOT will take into account the research work carried out all over the world, and will be more realistic.

FINAL COMMENTS MADE AFTER THE FIAT TECHNICAL PRESENTATION

The technical presentation ended on a hopeful note on future regulations. A quick look at the new 3,000 pound ESV Specification released by the DOT shortly before the Fourth Conference would not seem to justify these hopes. In fact, the Specification requirements reflect very little of what has been accomplished in the different countries since the international research program was launched.

The changes in respect to the first 4,000 pound ESV Specification are limited to a modest recognition of the existence of the compatibility problem, and to a modification of some requirements which were marginally important, like low-velocity impacts, or were manifestly excessive, like front and rear-end collisions against a pole. On the other hand, the severity level of other tests has been raised in spite of the serious difficulties that had already become apparent at the previous severity level — particularly the side impact, both against a pole and in car-to-car collisions. However, we will keep on hoping until September of this year when the final DOT Specification for an optimized family sedan should be released.

If the September specifications are in line with the present ones, we must come to the unfortunate conclusion that the huge amount of work which has been carried out so far by manufacturers all over the world has been of very little use. What is even more serious, any future work would also be of little benefit. In such a case, we would be forced to become passive bystanders and would have to accept the imposed standards. There would be no need to seek our collaboration; our mere obedience to laws would suffice.

We think that after two years of work since the start of the common program, the different studies conducted by the industries have already given sufficient information and results to set new goals which take into account not only the technical, but also the economic and social realities.

Briefly, our opinion can be outlined as follows:

- at the present state-of-the-art it is possible to meet, hopefully and in a relatively near future, the survival space requirements, provided the conventional severity level of the different crash tests is commensurate with the 30 mph impact for the front-end collision against barrier. Levels exceeding this speed would lead to technical and economic implications which, in our opinion, are not all proportionate to the benefits obtainable.

As far as crashworthiness is concerned, we have tried to demonstrate that the most important problem yet to be solved is compatibility. It is on this problem that the major part of all our work and studies should be focused.

- Another basic objective is the improvement of restraint systems which really give effective protection of occupants.

Let us try to reach these goals as soon as possible: we will thus render a great service to mankind. Any more ambitious program would at the present time be irrational and would represent only a waste of time and money.

We really have devoted a considerable amount of effort to the ESV program. But we should not forget that from a cost-efficiency point of view we have important social responsibilities towards our customers and the public opinion.

SECTION THREE

ACCIDENT AVOIDANCE SEMINAR

PART 1 INTRODUCTION BY CHAIRMAN

DR. MASAICHI KONDO, *Director*
Japan Automobile Research Institute
Japan

PART 2 STEERING, HANDLING AND BRAKING

PER-UNO UNOSSON
Saab-Scania
Sweden

S. TSUCHIYA
Toyota Motor Company
Japan

JOACHIM H. SORSCHE and DR.-ING. KURT ENKE
Daimler-Benz
Federal Republic of Germany

PAUL OPPENHEIMER
Girling, Limited
United Kingdom

HUBERT ALLERA
Citroen
France

JOHN ROSENKRANDS
General Motors Corporation
United States

FILIPPO SURACE
Alfa Romeo
Italy

B. INGRAM and PAUL OPPENHEIMER
Girling, Limited
United Kingdom

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DR. MANFRED BURCKHARDT
Daimler-Benz
Federal Republic of Germany

EGON-CHRISTIAN V. GLASNER
Daimler-Benz
Federal Republic of Germany

PART 3

VISIBILITY, LIGHTING AND DRIVER ENVIRONMENT

KIYOKAZU YOKOI
Japan Automobile Research Institute
Japan

MR. HAMMERSTEIN
Adam Opel, AG
Federal Republic of Germany

PART 4

CONCLUDING REMARKS

DR. MASAICHI KONDO, *Chairman and Director*
Japan Automobile Research Institute
Japan

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE

PART 1

INTRODUCTION

DR. MASAICHI KONDO, *Director*
Japan Automobile Research Institute
Japan

Good morning, ladies and gentlemen. I am to act as the Chairman for the seminar on accident avoidance. My name is Dr. Kondo. I serve as the Executive Director of the Japan Automobile Research Institute, whose acronym is JARI. Based upon the conferences we had in Paris and Stuttgart, we decided to have this seminar in the form of a panel discussion.

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

PER-UNO UNOSSON
Saab Scania
Sweden

I would like to take up some matters concerning mainly the newly issued ESV specifications and address some questions, especially to the DOT members, to see if there can be fruitful discussions around these matters. It's very essential for safe driving that handling performance doesn't change abruptly during different conditions in driving modes. We've heard about this yesterday, especially from Fiat, and this is apparently well considered in the issued ESV specifications. Steering response and stability criteria are well covered for constant velocity modes, but criteria must also be established for conditions during acceleration and retardation. The handling techniques of the vehicle must not change so much that safe driving will be jeopardized. In contradiction to what Mr. Scott said this morning, I will raise the question: Is it not, from a safety point of view, better to have a car with a small engine than with a too-powerful engine? The behavior of an underpowered vehicle during acceleration is predictable; with an overpowered vehicle, however, an ordinary driver can abruptly lose control of the car when available horsepower is used in a critical situation. I would like to emphasize my statement by calling your attention to Figure 1.

Assume a situation where the driver will accelerate and simultaneously steer to follow a curved road or to avoid a car which he is going to pass. Assume further that a moderate, front-wheel angle of one degree is established by the driver. If the speed is around 30 mph and the road surface is dry, that means a road friction coefficient of about .8; the vehicle is under control. If the road is covered by snow or ice, the μ is less than .4 and the vehicle suddenly goes out of control and into the out-of-control region. The longitudinal acceleration (a_x) value of .15g, shown, represents the minimum average acceleration required in the ESV specification for acceleration from 30 mph to 70 mph. For the same acceleration and steer angle, the vehicle is out of control at 70 mph if available road friction is .8. My question is, shouldn't the maximum engine power be specified as well as, or rather than, the minimum power? I ask the chairman if there are some comments on that before I proceed.

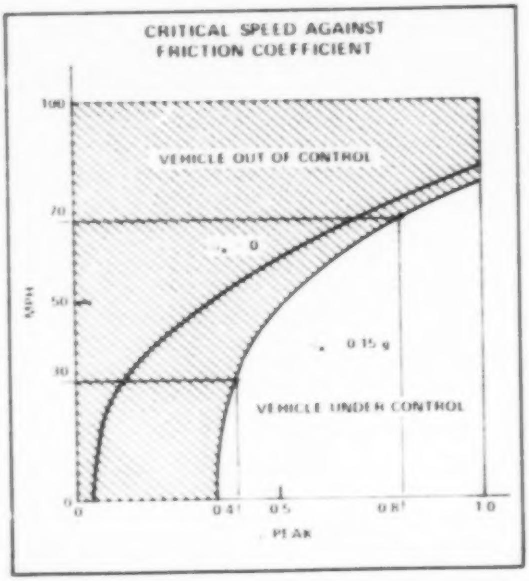

Figure 1

Mr. DiLorenzo: I think there's a good point here. In the hands of an inexperienced driver, an overpowered engine can get him into gross difficulties. When we originally wrote the specification, we were concerned that because of the weight problems in the ESV, we would end up with a very small engine. We felt that the likelihood of getting a car that was grossly underpowered was far greater than getting one that had too much power, and this has indeed turned out to be the case. Some of them, I don't recall exactly which ones, didn't meet the 12-second acceleration from 30 to 70; it has been difficult to obtain. But going back to this discussion, I think we should reconsider this in light of the fact that there are certain power plants now, and I think in particular of the Wankel engine, which do offer a high-horsepower capability for a very small volume and, indeed, we could get into a situation that has been described here.

Mr. Unosson: Thank you. There is another matter that has been touched on, especially from the Mercedes people, and I would like to emphasize it. In existing ESV specifications, tests involving the vehicle dynamics are to be formed by a skilled test driver. Road safety, however, is, to a great extent, dependent on how the car is controlled by an average man. The driver-vehicle system has to be matched in an adequate way. I'm not convinced that the vehicle marginal performance when driven by a skilled test driver is the best measure of road safety. Vehicle behavior in normal modes, when driven by an ordinary driver, ought to be more adequate. Such important items as steering overall ratio and maximum steering wheel torque are not specified in a direct measurable way. These parameters are related to the strength of a skilled driver and how fast he can react in a critical marginal situation and disregard what an ordinary driver can accomplish. My question is, shouldn't repeatable and directly-measurable test procedures and requirements for steering ratio and torque be specified and adapted to the ability of an ordinary driver?

Mr. DiLorenzo: Our requirements for steering torque were based on cars that had been produced and that the average drivers were used to. I do feel, though, that there is a necessity to give the driver more feedback, so to speak, so he can better understand his car and evaluate what's going on. We do need research to decide on just what this feedback torque should be and how much. As for inputs, or how fast we put steering into the vehicle, we have tried to employ wherever possible in the ESV specifications open-loop responses which have no feedback by definition. In this respect, we wanted to eliminate the variability of steering input. So, we did do some work on defining how fast a large driver or strong driver could input steering, and we based our open-loop responses on that. And it turns out that the rate of input, 500 degrees per second, is humanly achievable. Since this rate of input is beyond what the car can respond to, it eliminates that variability.

Dr. Kondo: If the average driver is in an emergency situation, what kind of reactions will he make? I think this is the kind of task that we have to address in research and this is the kind of topic that we really have to tackle. Now, Mr. Unosson please.

Mr. Unosson: I would like to take up the question of braking, especially the design of brake circuits, the splitting of the different brakes. Splitting the brake circuits front and rear—we call it black and white—is encouraged in the ESV specification. In my opinion, it's not advisable even if it could, to some extent, be justified by anti-skid on the rear wheels. One must not expect a front system failure together with an anti-skid failure. The car, however, is very easily out of control with a front system failure, even if the rear wheels are prevented from locking during braking. During braking, when the front brakes are not functioning, the steering ability of the free-rolling front wheels is good. In a critical situation it's most likely that the driver will take advantage of this, for instance, to avoid an obstacle.

Take, for instance, the following conditions shown in Figure 2. To avoid a critical situation, the driver makes a moderate steer angle so that the steering wheel is moved out one degree; the anti-lock system works ideally on the rear wheels and regulates them to a slip value under that which causes the peak of the μ slip curve. However, if, under these circumstances, the driver brakes at 60 mph with a retardation of .2g, the whole central axis and the surface gives a grip so that the peak μ value of .8 can be achieved and the car is under control; however, going up to the specified .31g deceleration, the vehicle is suddenly out of control. On a slippery road, for instance, with μ less than .3, the zone of safe operation is limited to very small retardations and low speeds. With this as a background, we can summarize as follows: with a brake circuit failure, the driver loses brake performance, and there's nothing to do about that. Also the simultaneous loss in pedal stroke can confuse the driver in the situation. Under these circumstances it is not advisable to decrease the stability and maneuverability of the vehicle. My question is: should black and white brake line splits be allowed?

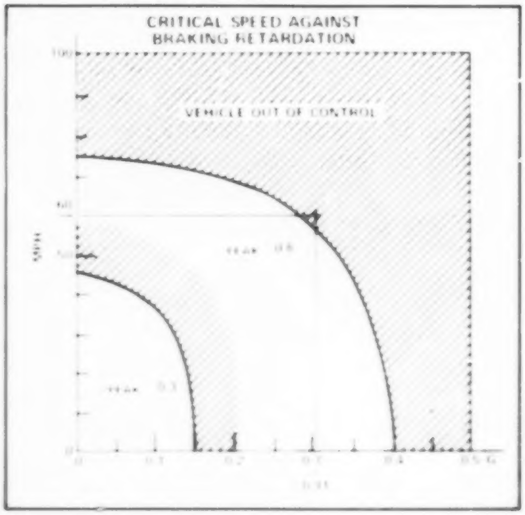

Figure 2

Mr. DiLorenzo: In our revised brake specifications, if I understand Mr. Unosson correctly, the black and white split is the front and rear axle?

Mr. Unosson: That's what I meant.

Mr. DiLorenzo: We didn't intend that to be a design limitation. In other words, the fronts are tied together and the rear axle is tied together. In fact, we would encourage, if it can be justified by design or performance, that you choose whatever hydraulic split you feel is the right one. With respect to the pedal motion, the change in pedal motion due to hydraulics, I'm not sure I have any comment to make on that, other than we do require that the driver's controls take into account the 5th percentile female to the 95th percentile male and that in that consideration the pedal travel will not be excessive or beyond what the smaller driver could retain.

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

S. TSUCHIYA
Toyota Motor Company
Japan

At the present time, I am engaged in experimental research at the Higashifuji Technical Center on the maneuverability and stability of vehicles.

Basic ideas for formulating specifications to develop ESVs, with particular emphasis being given to the prevention of accidents, include the following:

1. Priority should be given to potential techniques during testing. At that time, no attention should be paid to problems that may develop in the mass production of test vehicles;
2. During the construction of an ESV, factors which may result in accidents should be controlled so that they conform to feasible, specified conditions;
3. A study should be made to determine whether the specifications adopted to standardize the objectives are appropriate and to determine pre-accident safety requirements for the future;
4. A determination of safety approaches, and their application, is necessary to raise the pre-accident safety level in the motor vehicles of the future.

Consequently, the ESV specifications are considered only one of the ways in which safety may be attained. With this in mind, we set up the study not merely to meet specification requirements, but also to build trial cars with performance characteristics that went beyond the specifications. In other words, the experimental vehicle built under the ESV program is not an end in itself. Indeed, the goal of the program is the creation of safety tests and their eventual application to the cars which will be supplied to the public in the future. If such tests are not created and their results are not applied there will have been no value to all our work.

I do not wish you to feel that ESV specifications should be harnessed to the unknown quantity of future laws and regulations. Nor should you feel that they will be applied solely to mechanical, functional or other components of the automobile. Rather, the process of ESV development embraces political and mechanical experience as well as the successes and failures that may occur during the testing period.

It must be said, however, that it is difficult to set performance targets or goals and determine evaluation techniques. Steering and handling present special problems. Let us look at braking performance as one example of the problems that may arise. It is clear that the shortest braking distance and smoothest action are desirable. However, at this stage, it is not possible to reach the maximum desirable goal. Another example may be found in Volvo's presentation. Yesterday, Volvo's representative mentioned that difficulties arise when a negative caster offset is used.

Although side-wind stability was assured during course deviation, steering wheel damping deteriorated. As a result, the driver felt less secure. These and other findings mentioned yesterday coincide with my own opinions.

I should like to take this opportunity to add a few other comments. During the second phase of American testing, numerous changes were made in the specifications so that the result obtained was within the acceptable range. However, it should be noted that the yaw response in the Japanese vehicle was found to be on the neutral steer side rather than in the acceptable range.

Let me return to the subject, that is, good performance in relation to the maneuverability and stability of a vehicle and make the following two points. In the first place, in addition to physically measurable performance, the driver's feelings are extremely important. It is critical, of course, to maintain stability under accident conditions. However, under normal conditions, it is necessary to avoid anything that gives the driver an increased psychological or physical burden which might cause fatigue. In other words, anything that may give the driver a feeling of discomfort or resistance must be avoided.

Secondly, the influence of the environment affects good performance to a tremendous extent. Among the major environmental factors that must be considered are highway conditions. Consequently, driver reactions should be obtained before environmental factors and vehicle design are modified. It is natural

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to expect that the driver would like to operate a vehicle which is adaptable to all environmental conditions. He would like to have the performance features of his car change as little as possible regardless of changes in the environment. In this sense, I feel that we should develop a car which will be both easy to drive and which will maintain stability even under both normal and adverse conditions.

Please excuse me for making some abstract comments in the course of this presentation. A more detailed analysis will be presented during the discussion sessions which will follow this meeting. However, it is my hope that this brief report has provided you with an outline of our work.

Dr. Kondo: I wish to emphasize to the foreign participants that Mr. Tsuchiya's statement included the following: in Japan the vehicle driver system characteristics are very important and this recognition has existed in Japan from an early period and SAE Japan and JARI had a joint committee to conduct a feeling test for the drivers. That is, this committee was assigned to test what kind of feelings and comfort drivers get out of driving vehicles; a portion of this test has already been completed. This I wanted to introduce to you.

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

JOACHIM H. SORCHE
Daimler Benz
Federal Republic of Germany

My statement consists of three parts. First, basic remarks to the whole subject of accident avoidance. Second, some remarks concerning brakes, especially the new specifications for the intermediate ESV and third, some remarks presented by a colleague of mine, Dr. Enke, concerning handling.

First, accident avoidance, as a whole. The subject, accident avoidance, which we are discussing today is indeed extraordinary in importance for safety on the road. Unfortunately, the subject is also extremely complex. Therefore, we should be most careful in such discussions and in rule making. The areas of standards whose consequences can easily be comprehended for instance, color and size of signal lamps, stopping distance from a given speed, and similar items have long since been exhausted. In such instances as these, benefits and possible drawbacks of regulations have been apparent to everybody concerned. The field of crash injury reduction, on which much attention has been focused in the past years, is becoming more difficult to assess. However, the cost-benefit analysis permits a reasonably effective evaluation and the disadvantages can be readily recognized.

In the field of accident avoidance, however, the situation is basically different. In this area we frequently do not even know whether a requirement imposed for a specific aspect of vehicle handling would improve the performance of the driver-vehicle combination. It is even more difficult to foresee the disadvantages for other driving disciplines or to avoid them. For instance, increased requirements regarding direction and stability might entail a poorer steering response. In the same way, upgraded specifications for braking efficiency might cause an impairment of brake stability that would cause breakaway during a stopping maneuver; or augmented demands on stopability, in the event of failure of one circuit, will mean distinct disadvantages for the service brakes. In the same manner, the consequence of increased angle of visibility may be a reduction of roof stiffness, the installation of very wide tires which give high cornering forces can make it difficult to drive straight ahead on a road with longtitudinal grooves and increase the danger of aqua-planing on the road surface, and so on. We feel that the hasty creation of regulations without sufficient prior consideration and thoughtful deliberation of all imaginable disadvantages will uselessly limit the design freedom of the engineer and thus result in a less-advantageous compromise for the total vehicle. It is for the above reason that one should proceed extremely carefully in the delicate field of accident avoidance. Only issues whose necessity and possible drawbacks are clearly recognized should be regulated and then only when absolute agreement as their acceptability for all designs has been reached. Above all, it should be kept in mind that the present state of the art of design for accident avoidance has, in fact, been reached without regulations. In this area of accident avoidance we are, however, close approaching true physical limits imposed by the tire-to-road contact relationship when we consider the performance of many cars which today represent a good design compromise.

Now some remarks concerning brakes. It's our opinion that the majority of the present brake systems are not so bad, that extremely complicated regulations have to be issued, the usefulness of which may be doubtful. Standard 105 and 105a, as we see it, still represent an acceptable compromise in view of the number and complexity of the specifications, disregarding, however, all data and particulars. Regulations establishing still more complex details are, therefore, a disadvantage. This is more so since, fortunately, vehicles of quite different concepts are built all over the world, and these, naturally, have different advantages and drawbacks. It is hard to imagine that more detailed regulations shall result in an equally favorable layout compromise between advantages and disadvantages for all of these differing basic concepts. The brake requirements detailed at present by the new technical specifications for an intermediate ESV surpass by far in this respect what we can advocate. In addition, the specifications apparently have been issued rather hastily and without the necessary careful examination; otherwise, some of the physical contradictions could not be explained. Since it would lead us nowhere to discuss the technical specifications in every detail, a few items may suffice. First, we cannot agree with the procedure for evaluating vehicle brake performance using the requirements deducted from the Highway Safety Research Institute report. We feel these requirements are too demanding and foresee the disadvantage that even if one would succeed in realizing the requirements by an unduly high technical effort, this would be possible only with vehicles which display unstable performance in straight forward and curved retardations. The total brake efficiency theory is based upon the fact that the distribution of the brake forces on the front and rear axles should, if feasible, be closely proportionate to the dynamic wheel loads. However, we all know that a better utilization of the coefficient of friction by the rear axle than by the front axle has considerably more severe consequences for the vehicle stability than vice versa. In other words, lock up of the rear wheels is extremely dangerous, whereas front wheel lockup is relatively inconsequential since the vehicle remains stable.

Secondly, we [illegible] that the calculation procedure employed for the ESV, as we see it, applies only to four wheel anti-lock systems, while the requirements of the technical specifications demand a two wheel anti-lock system as a minimum. So, we feel that the values specified here for the brake rating of those vehicles on which the investigation of the Highway Research Institute is based, are not complied with. In our opinion, the procedure is not only unnecessary but also disadvantageous.

Thirdly, we have strong reservations as to the ESV specifications as well as to the existing extent of 105 and 105a when in both instances, lock up of a front and rear axle is considered to be equally dangerous and, therefore, prohibited. We tend to reject the requirement that in all brake applications the driver should obtain the shortest possible stopping distance without being permitted to lock the wheels. In this way all of the stopping distances compared are longer than actually obtainable with the given vehicle. This evaluation procedure, therefore, is meaningless and the driver encounters a problem which is almost impossible to tackle. The extent to which this fact has a bearing on the result is easily shown by a comparison of the consumer information in which the stopping distance of the best and poorest vehicles have a ratio of 1 to 2, or 1 to 2½, even though in this normal straightforward brake test, the real stopping distances of all these vehicles should lie rather closely to each other.

Fourthly, we understand the desire to regulate stopping distances in a curve, but the specified stopping distance will not match the respective deceleration. Apparently, pressure build-up time has been omitted. The specified deceleration values could be met, but not the required stopping distances in all cases, even when using a good anti-lock system.

Basically, we would welcome a mutual effort in order to come to performance requirements instead of too many design requirements. This is amply documented by the required regulation procedures, exclusion of wheel lock up and the like. We feel that it is more significant in a given situation to have as short a stopping distance as possible without loosing control and without leaving the lane in straightforward or in curved roads.

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

DR. ING. KURT ENKE
Daimler Benz
Federal Republic of Germany

To develop standards in the fields of steering and handling we must work in one of the toughest areas. Roughly speaking, we have to consider the following points:

1. Response of the vehicle to the steering input from the driver and to influences from the road and meteorological conditions.
2. Actions of the driver as a consequence of his intentions and sense impressions.
3. Transmittal of such impressions by the environment and vehicle.

In the ESV specifications, as in the majority of all research conducted so far, only the vehicle response has been covered. The second and third fields, which without any doubt most heavily influence and are a major decisive factor in traffic safety, have barely been scratched. For instance, we absolutely do not know what percent of all drivers utilize the handling qualities built into their vehicles and which of these properties are specifically important for any given emergency situation or whether they are important at all.

It is unknown whether the average driver operating a vehicle equipped with an antilock system will, when confronted with an impending rear end impact, act correctly by applying the brakes and circumventing the obstacle, or whether he will step on the brakes, firmly hold the steering wheel and do nothing. Much more statistical data will have to be accumulated, which certainly will not be a low-price project.

Another problem quite often overlooked is the immense extent of the field of vehicle handling which makes it impossible to categorize the most influencial vehicle characteristics in just a few tests.

Figure 1, for example, gives you a compilation of the most frequently used steering maneuvers listed from the very simple to the most complicated. Next to the description you will find the vehicle path and the path curvature given in curves and symbols.

Figure 2 shows also in curves some basic possibilities of the accelerator and brake pedal positions.

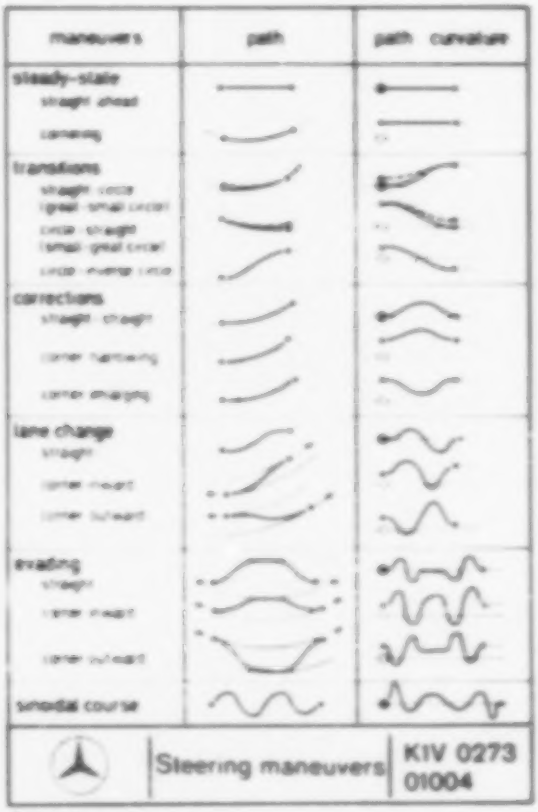

maneuvers
path
path curvature
steady-state
straight ahead
cornering
transitions
straight-circle (great-small circle)
circle-straight (small-great circle)
circle-reverse circle
corrections
straight-straight
corner narrowing
corner enlarging
lane change
straight
corner inward
corner outward
evading
straight
corner inward
corner outward
sinoidal course
Steering maneuvers
KIV 0273 01004

Figure 1

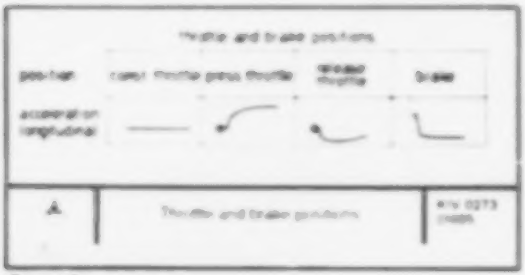

throttle and brake positions
position
const. throttle
press throttle
release throttle
brake
acceleration longitudinal
Throttle and brake positions
KIV 0273

Figure 2

Figure 3 illustrates a combination of both: steering and accelerator or brake maneuvers which certainly can be performed at the same time. We don't believe

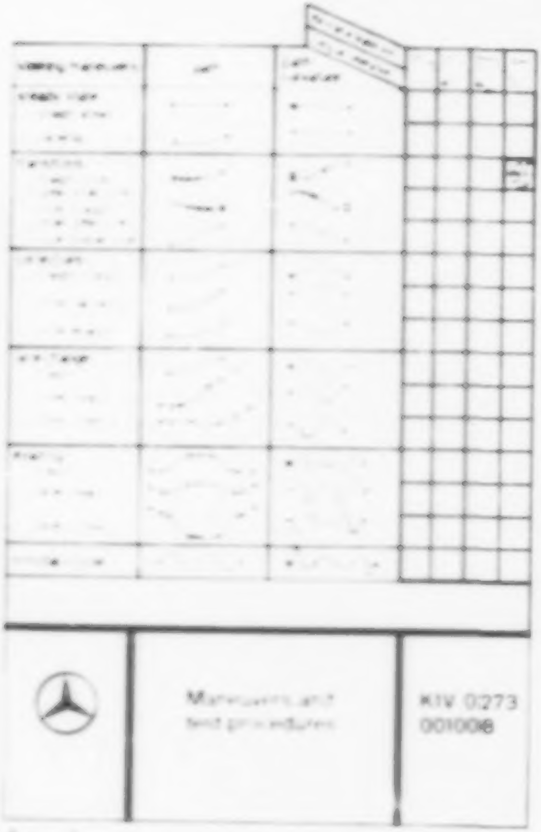

Maneuvers and
test procedures
KIV 0273
001008

Figure 3

that we can disregard even one of these maneuvers. As a result we have a grid with [illegible] squares.

Figure 4 shows one of these squares through a magnifying glass. Within the magnified area shown, on the top left side, is indicated the amount of driving maneuvers with respect to variations of driving speed and corresponding quickness of the steering input. In the left lower corner are listed some of the data to be accumulated. In the upper right square is an illustration of the plotting and comparison of evaluation results. In the lower right square these evaluation results have been combined with the human reaction, which as yet we know very little about.

Figure 5 shows the amount of research to be conducted, when we attempt to grasp the most important operating conditions. You will agree that this is a mountain of a work program a sheer nightmare for the fellow in charge who has to decide which areas to cover and which to drop.

It is not a basic dislike of rules and regulations which establish our pessimistic point of view, it is our lack of knowledge in the flexible control parameters

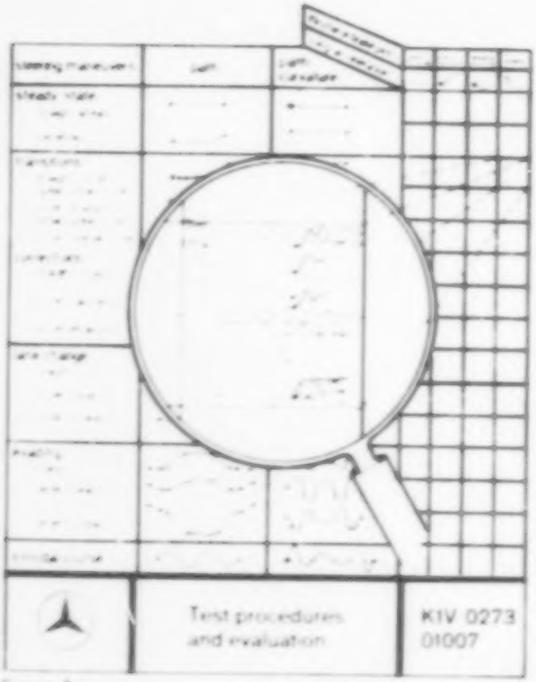

Test procedures
and evaluation
KIV 0273
01007

Figure 4

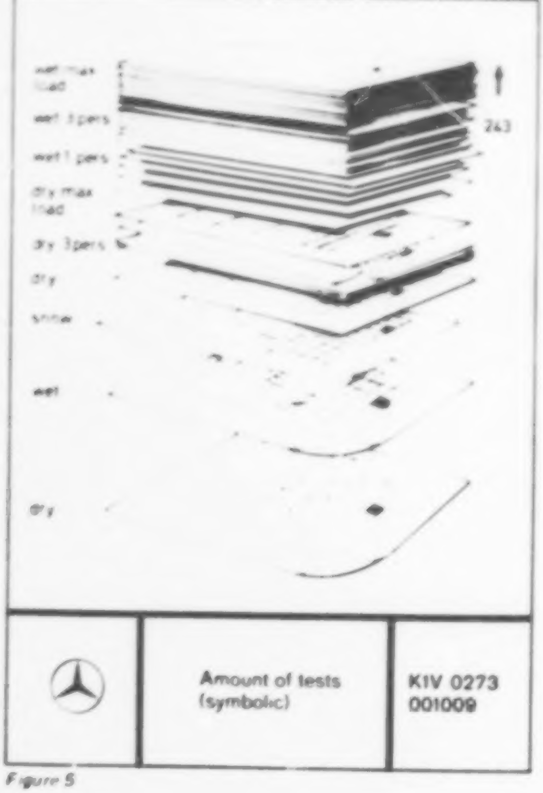

wet max load
wet 3 pers
wet 1 pers
dry max load
dry 3pers
dry
snow
wet
dry
243
Amount of tests
(symbolic)
KIV 0273
001009

Figure 5

of the human driver. If comparison tests resulting in objective data without the test driver influence could be made we would be very happy, since this would permit management to base a decision on data without subjective judgement.

Priority should be given to the middle and upper acceleration range, because evaluation will show whether one may even come close to an actual emergency situation.

We are still of the opinion that in order to gather the most meaningful data, tests with test drivers should be run, because they are capable of performing a series of maneuvers in one test run and evaluating them.

That's actually how industry handles it today. In our case, for instance, experienced test drivers test our vehicles under given environmental conditions between the north of Sweden and the middle of the African deserts. Only in this way are we in a position to gain meaningful information on the qualities of any given vehicle.

Now, let me say some words about the handling requirements of the 3,000 pound ESV.

After considering the variety of handling maneuvers and conditions required, they can at best represent only a very small portion of the whole subject, and their relationship to actual safety is questionable. My comments concerning a particular test should, therefore, be understood only in this sense.

For steady state yaw response, the steering wheel angle multiplied by the steering ratio should be taken instead of the front wheel angle. Otherwise, the very important steering elasticity would be neglected.

In the lateral acceleration test, we would like to suggest that only acceleration values equal to or lower than those required for the basic layout be used, but no higher values.

The length of wind application in the crosswind sensitivity test should be considerably longer and should be performed with a driver, to also test the capability of the driver-vehicle combination in bringing the vehicle back to the original course.

Testing of the overturning immunity by the HSRI procedure is not acceptable. Only such procedures which can be carried out by a human driver should be used.

What we really need, but do not have today, is information about the relationship between handling and accident avoidance that is based on the investigation of real accidents. But listening to drivers and what they tell us will not be enough; we need true-time histories of steering inputs and pedal movements.

Mr. DiLorenzo. I want to first thank the Mercedes-Benz people for their very thorough paper and I would like to comment on Item 1 concerning the accident avoidance, and then perhaps a word or two on Dr. Enke's comments on handling.

As a general note on accident avoidance specifications, I would like to say that primarily because the progress in the area of crash injury reduction has been fast relative to accident avoidance, we have to do some things in accident avoidance to keep pace and ensure that we do have vehicles that people can handle to some degree. In fact, we have seen in the presentations this week that more and more goes into the vehicle structures and that these demands become more and more significant on vehicle handling. The primary intent of the ESV specification in accident avoidance and vehicle dynamics is not to guarantee safety from the standpoint that we know what safety is; it is only to assure that what we build, having been modified from existing machinery, does, in fact, behave like the vehicle we've been used to. This is especially true in our government procurement cycles where we don't have the staff of skilled drivers, and so forth, to assure that these very fine details are covered in vehicle dynamics; we have to assure that the vehicles we build do emulate current production cars.

I understand that because we use open-loop responses in our tests we are not including the driver, and this is because of our ignorance in that area. I think that's an area that must be pursued and I should state that our administration has on the street, I believe, at this time two requests for proposals to begin work in this area where we involve the driver and the vehicle dynamics. There are two approaches to the problem; they differ somewhat, but I'm optimistic at this point that as Mr. Scott mentioned briefly, we will be able to use some of these data in the 3,000 pound car program.

I would now like to comment on the point Dr. Enke made relative to the front wheel angle measurement. It is labeled that way on our curves but, in fact, we measure this angle from the center link of the steering mechanism as calibrated from a static position of the car. The steering wheel is turned and the front-wheel angle is measured statically and this calibration is put into a linear potentiometer that is actuated by the center link of the steering mechanism. So, although we have labeled it front-wheel angle, and term it front-wheel angle, it's actually the steering-wheel angle that's in there, and if there is compliance in the steering suspension during the maneuver, then this effect is in the specification.

Concerning the crosswind sensitivity test mentioned by Dr. Enke, I agree with him that it should be a longer test. But this requires an expensive facility and when we first developed our ESV specifications, we were limited to a degree as to what we could spend in this area. Our contractor was only able to come up with a 20 ft. exposure to wind, and that's why the specification reads as it is. The Volkswagen facility is more appropriate.

Concerning our overturning immunity test, I agree that drivers are the ones that overturn cars and not machines. But I have difficulty in resolving the fact that we do have a strong variance in drivers, even test drivers, and that if we want to be fair and test a number of cars and subject them to the same situations, we'd have to use something that is more repeatable. While this maneuver is now experimental, the reports on the HSRI work have just been released and we would like to try it in the ESV program and have included it as a specification. As we said this morning, it is a goal; this whole specification is a goal and subject to change as studies progress.

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

PAUL OPPENHEIMER
Girling, Limited
United Kingdom

I'm grateful for this opportunity to discuss the new U.S.A. Intermediate ESV specification which I will refer to as ESV 2. These notes are only my personal preliminary comments. I hope that you will accept them as constructive criticisms which will lead to a fruitful discussion from which we'll ultimately help prepare a basis for a uniform worldwide regulation, as suggested this week by Mr. Eckbert from Sweden and which may be applicable for mass-production safety cars as suggested by Mr. Wagner of Germany. My remarks will be confined to the braking performance requirements of ESV 2.

The proposed weight of ESV 2, is commonly referred to as a 3,000 pound car, but for brake engineers the gross weight of 3,800 pounds is more relevant, especially when assessing the pedal effort limitations in Figure 1 of the specification. This would be a relatively large car by European standards, but I believe such pedal efforts can be achieved unless there is a drastic reduction in the available vacuum for the booster due to further emission control devices. My first criticism concerns the use of the word deceleration. Your deceleration is based on the average value throughout the stopping period, but in practice a significantly higher deceleration is essential to cater to the buildup or response time of the normal power-assisted braking system.

May I suggest that your deceleration based on stopping distance might be called "mean" or "average" deceleration while the higher values be called "maximum" deceleration, or "fully developed retardation," which is a very clear definition used in the Swedish regulations. I hope that point is clear to you, Mr. DiLorenzo. Practical tests have shown the ratio of maximum-to-mean deceleration to be between 1.20 and 1.25 for normal braking systems. On this basis, the ESV 2 service brake performance requirement of 175 ft. stopping distance from 60 mph can be equated to a maximum deceleration of .85g, which provides a better appreciation of the stringent performance requirement, especially when we remember that the test must be carried out on a road surface skid number between 70 and 80. This also necessitates a very good tire performance which can provide peak friction coefficients well above the sliding friction coefficient measured with the ASTM tire. The respective maximum deceleration requirements for booster failure and front system failure are .5g and .38g, quite stringent requirements. They're all shown on the diagram.

May I, therefore, please ask for deletion of deceleration values when using stopping distance as a basis for braking performance or as a specific reference to mean or average decelerations to avoid ambiguity, also in Figure 1, which is particularly misleading.

My next point concerns the reference to performance testing on a skid number between 70 and 80. Would you accept the vehicle which meets the requirement on skid No. 80 but failed on skid No. 70, which is a very practical possibility? If not, then the range of conditions is misleading and a single value should be specified. For example, with reference to skid No. 75 which aligns with Standard 105a. I understand that NHTSA policy always states a single value, leaving the industry engineers to add their appropriate tolerances. To achieve compliance, the manufacturer would test just below skid No. 75 and NHTSA would test just above the skid No. 75. ESV 2's reference to booster failure takes no account of full-power braking systems such as we've already seen on several ESV's, notably those from Toyota and Honda. Please amend to power or power-assist failure.

Next, ESV 2 refers to front system failure, while the front-to-rear, or vertical, or black-and-white brake system split is the most widely used; it's not considered to be the safest solution. It is particularly deficient in the potential deceleration, especially on front-wheel drive cars which are popular in Europe and have other safety advantages. Please amend to hydraulic circuit failure, thus permitting the established advantages of more complicated splits which, in fact, are probably essential to providing almost .4g maximum deceleration in the failed condition on a skid No. of 75.

I would have liked to say more about your method of assessing braking performance by means of the factors E, F, R and M. May I just say that an E of 80% in conjunction with the stopping distance requirement of 175 feet appears to imply a maximum deceleration of about 1.07g. This is difficult to reconcile with a skid number of 70 to 80, especially since the only published results that I've seen of such measurement as shown by Mr. DiLorenzo at Stuttgart and by Mr. Murphy at Fisita, showed the values of E dry to be between 65% and 68% for two different cars with four-wheel, anti-skid systems, which is well below the required 80%. I was surprised to hear yesterday that the U.S.A.'s ESVs have achieved E values in excess of 80%. Are we sure that we're all talking about the same braking efficiency measured in accordance with the reference HSRI reports?

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

HUBERT ALLERA
Citroen Automobiles
France

My report is on the state of progress in the development of antilocking devices, limits of their performances and the difficulties of finding industrial viability. During the presentations which we have attended, there have been many questions on antilocking devices. We do not intend to doubt the progress that has been made by these many specialists, but I would like to try and make a quick, a very quick commentary on this. First of all, we must consider the problem of choice, the practical limits of possible braking improvements and the difficulties of the development in the industry of cars. Before taking up the study of antilocking devices, the engineers should choose both the type of car, commercial or prototype, and the different technical orientations for braking elements, whether drum or disc for the front brakes. The choice is particularly easy for high-performance European vehicles. Most of these vehicles have standard disc brake equipment. For the back wheels we think it would be practically necessary to give up the drum brakes. Should we keep the master cylinder or adopt a source of high pressure type full power? As far as we are concerned, the presence of a hydraulic control of high pressure used for the suspension of our cars is a question which we do not even need to go into because it has been proved to be the correct solution. Having considered all options, therefore, for the braking elements, we should then determine whether to limit ourselves to a rear axle antilocking system or make a complete equipment integrating the front axle. To this day we have adopted the most successful solution, but I must say that the action of the wheels poses very difficult problems; where the antilocking device is used, we cannot ask the driver to make corrections in steering to correct unsymmetrical right and left braking deformations. We believe that in the first stage there is a necessity to accept control by axle to avoid different braking power. Even in the absence of a reaction from the steering, because of a slip angle taken by the tires to balance certain jerky movements, we would like to give priority to stability and handling, and for that the axle action will be directed by the wheel which is in contact with the ground with a weak grip. We also give priority to the rate of slipping of the road tire. The antilocking device should resolve all the problems of pressure on the braking system. A too frequent action on the axle itself would be very dangerous for the driver himself.

As for the study of an antilocking device, we have asked for assistance from outside companies to help us study this problem, particularly in endurance tests. First of all, it was clear enough that if certain gains had been made on stopping distances, these gains were not very noticeable, except in certain cases for the driver himself. When antilocking devices are used they depend upon the tire road coefficient of friction that is, with good surfaces, they can be better controlled. So it is a question entirely of friction coefficient as to whether the braking will be conventional or whether there will be difficulties, but we must not forget the conduct of the driver under the different surface conditions. When the coefficient is very low and the driver is not very experienced, our tests are not very significant. We have found that spike tires, for example, could be used in any of this type of tests. We also have to take into consideration the fact that stopping distances vary with the different surface conditions, such as the presence of sand or melting snow, but these conditions do not always exist in France. We have heard that the locking of brakes in the last few meters of braking is also an extremely pressing problem that should be taken under consideration, as this could avoid many accidents. In conscience, having only taken up this very vast problem very superficially I would, however, like to end by saying that to assure the viability of an antilocking device, we will need to make many further studies and find perhaps even new devices, metallic type or others. On the other hand, since the manufacturing pace would only be a few units per day, this would certainly raise the price of the cars by at least $500 per car. For standard models I think that prices could be reduced a little, but, they would still be much too high. A lot of work has been done, but there is still a lot to be done and we are aware of this and continuing our research. We do not exclude the idea of a temporary failure, but we do hope that with the cooperation of the many industrial firms and others, we will arrive at some solution.

STEERING, HANDLING AND BRAKING

JOHN ROSENKRANDS
General Motors Corporation
United States

When the subject is accident avoidance it is tempting to talk about all the factors that may influence vehicle control, such as visibility, lighting, accelerations, control locations, braking, and pedal feel. The problem is that both driver, vehicle and environment influence each of these items and it is this interplay that constitutes vehicle handling. Several of these factors have relevance mainly to comfort and convenience, but common to all is that very little is known about their true relevance to safety, and this in my opinion is what the ESV must be about.

When General Motors delivered the ESV cars to our Government, we stated that no new technology was needed to provide the vehicle response characteristics specified in the contract. We also emphasized that although several production cars already come very close to the specifications, any relationship between these specifications and highway safety should not be assumed. On the other hand, I believe that the many millions of car customers over the years have influenced the design through a natural selection process. In fact, the limits of customer acceptability are likely to be considerably narrower than the limits of safety.

The question of what constitutes "good handling" has been discussed many times and in many places. The term "good handling" means different things to different people, and particularly for different traffic and road conditions. Personally, I will choose one type of car for a short trip where I have no problem staying alert. In fact, I enjoy the typical sports car for this type of driving. But for long trips I prefer a car more conducive to leisurely driving, which means a car that requires less constant attention. However, I like to think that I am equally safe in either type of vehicle. Differing viewpoints are often most obvious during international gatherings, such as the ESV conferences. I also wish to refer to the work going on in Subcommittee 9 of ISO's Technical Committee 22, where an attempt is being made to coordinate vehicle handling tests.

At the ESV meeting in Stuttgart, Mr. DiLorenzo spoke to the question of open loop vs closed loop testing—that is, with, or without, driver interaction. I wish to share with you some of our experiences in this area. But it is appropriate first to consider briefly some current trends in the United States.

Over the last seven years, the average highway speed has increased 10% (Figure 1). This means that vehicle kinetic energy in any given situation has increased approximately 20%, bringing about a corresponding increase in stopping distance or, in the event of loss of control, off-road excursion. All other things being equal, one might expect more accidents and accidents of greater severity. The fact that we have not seen such a dramatic increase in these respects is an indication that the driver-vehicle-road system is being improved. Although it is impossible to assess the exact areas of improvement, there is evidence that the vehicle, insofar as its handling qualities are concerned, has been improving.

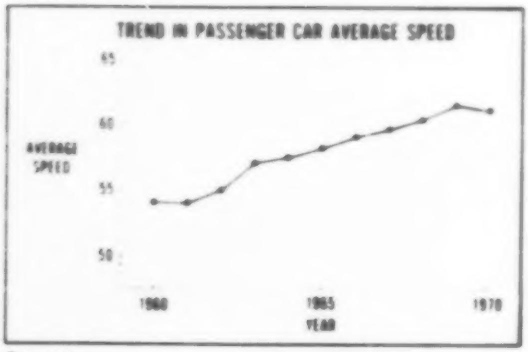


Figure 1

There has been a trend in power-assisted controls for greater application of power steering and power brakes (Figure 2). The trend in power brakes persists, although power steering is leveling off due to the rapid increase in small vehicles where it is not required.

Tires also are being improved (Figure 3). Our data indicate that the braking traction capabilities of our tires under wet conditions have improved appreciably over the past 10 years. Introduction of the bias-belted

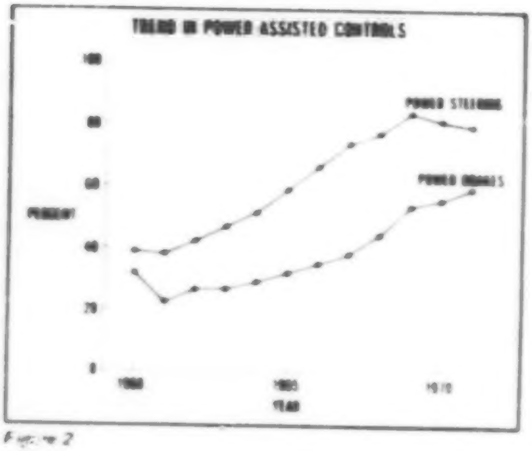


Figure 2

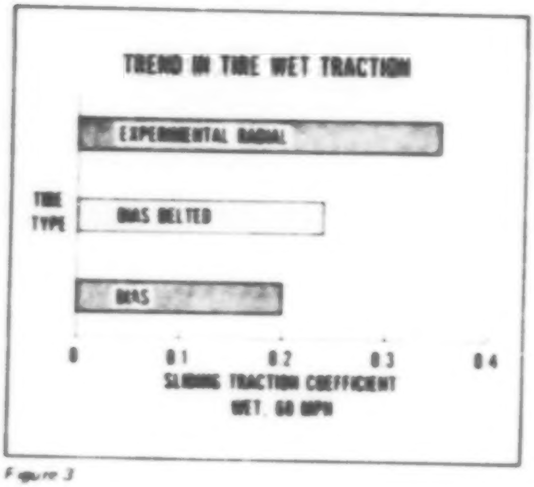


Figure 3

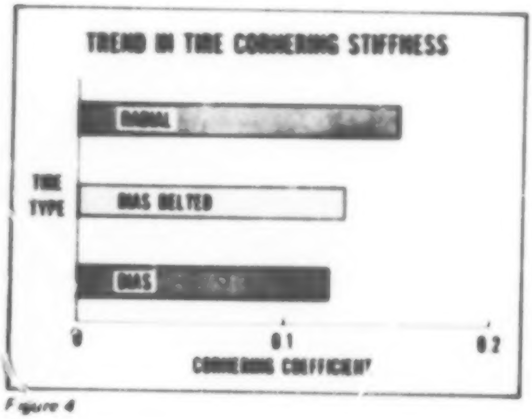


Figure 4

tire afforded a 20% improvement, and there is even greater potential for the radial tire, as indicated by data from some of our experimental designs.

There is evidence, also, of improved cornering capabilities in tires (Figure 4). One of the several important tire characteristics is that of cornering stiffness which has steadily increased. We can do well in this respect with radial tires and our problem of tomorrow may not be in improving the state-of-the-art of handling-related tire properties, but in insuring that all tires, both original equipment and aftermarket, will meet equivalent performance levels. There is, therefore, reason to believe that vehicles have, and are improving, in respect to the more stringent traffic system demands.

To date, a number of proposals for vehicle handling tests have been considered (Figure 5). Some of the more important contributions are the results of work sponsored by the U.S. Department of Transportation, NHTSA. The Highway Safety Research Institute at the University of Michigan has proposed vehicle-response tests which stress high-lateral acceleration-limit control performance and exclude the driver influence. However, in their report in November of 1972, HSRI did not comment on the quality of such tests, stating, "much research is needed to provide an understanding of those factors which determine driver-vehicle performance under emergency conditions. Specifically, experiments must be conducted which can demonstrate the relationship between driver-vehicle system performance and the open-loop properties of vehicles such as have been measured in this study." The Cornell Aeronautical Laboratories which now is called CALSPAN, has proposed a series of task performance or closed-loop tests. The experimental safety vehicle tests, which originated from DIGITEK, cover a broad range of dynamic response tests and include some aspects of task performance. However, a connection with accident causation circumstances has not been established except subjectively by the contractors. The relationship of the dynamic response tests to safety has been particularly difficult to establish because of their very technical performance measures and the problems of relating these difficult-to-

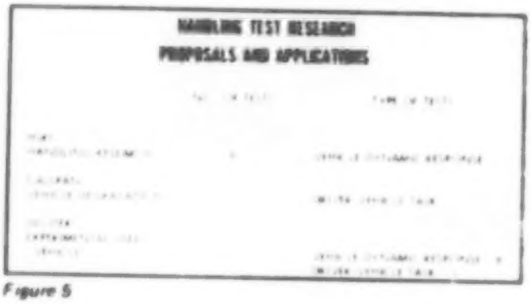
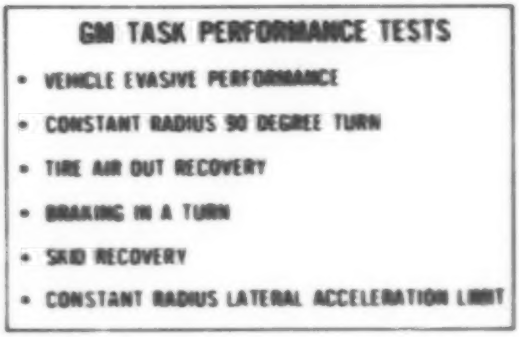

HANDLING TEST RESEARCH
PROPOSALS AND APPLICATIONS

Figure 5



GM TASK PERFORMANCE TESTS

- VEHICLE EVASIVE PERFORMANCE
- CONSTANT RADIUS 90 DEGREE TURN
- TIRE AIR OUT RECOVERY
- BRAKING IN A TURN
- SKID RECOVERY
- CONSTANT RADIUS LATERAL ACCELERATION LIMIT

Figure 6

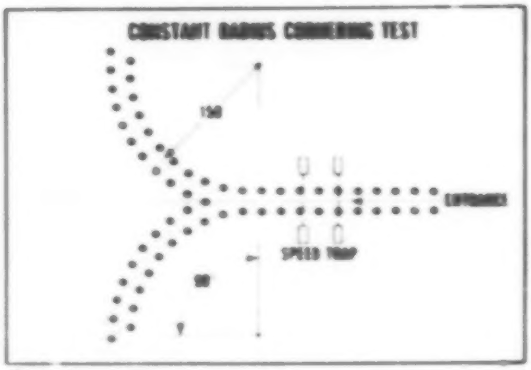


Figure 7

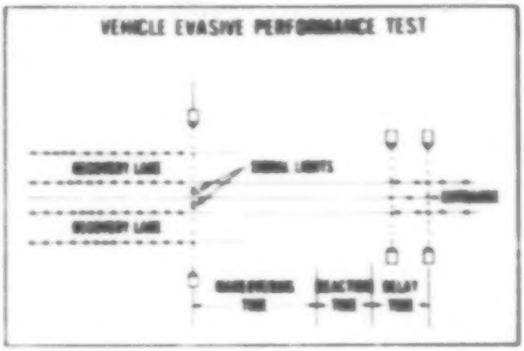


Figure 8

understand measures to vehicle path-following capability when the vehicle is under driver control. The greatest deterrent to establishing the relevance of these tests, however, is probably the lack of good accident-causation information.

In the absence of such information, General Motors has proposed a number of tests that subjectively appear to have value (Figure 6). These tests involve both the driver and the vehicle, and have an objective measure of performance. Some of the situations that have been proposed for study are: vehicle evasive performance, highspeed transient cornering, tire blowout, braking-in-a-turn, skid recovery, and steady-state cornering capability.

An example of a task performance test is the constant radius ninety degree turn test. This simulates a situation where a driver enters a turn at an excessive rate of speed and is likely to exceed the cornering capabilities of the vehicle.

The test course is laid out with traffic cones and defined by lane width, turning radius and angle of turn (Figure 7). The criterion of performance is the maximum entering speed obtainable in succeeding runs.

Another test of a similar nature is the evasive performance test (Figure 8). The purpose is to measure the lane-changing capability and recoverability in a maneuver that requires a 12 foot lateral displacement. You enter the speed trap at a prescribed velocity, and a signal light tells you which of three lanes to choose. The timing of the signal light can be varied, and the criterion of performance is the minimum maneuvering distance after the light comes on.

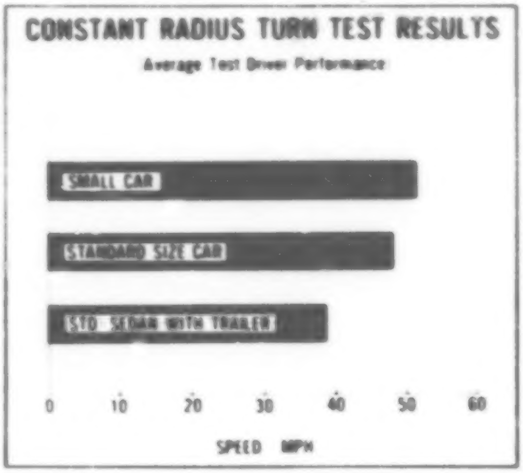


Figure 9

These tests have been studied extensively at General Motors (Figure 9). The data for the constant radius-turn test indicate good correlation with subjective opinion for such a maneuver. There is little difference between vehicles except when a trailer is added. The reason that the small car can negotiate the turn slightly faster than the standard size vehicle is that the same lane width is used in all the tests.

The results from the evasive performance tests indicate the same trend as the constant radius turn tests, both in regard to the influence of vehicle size and to repeatability (Figure 10).

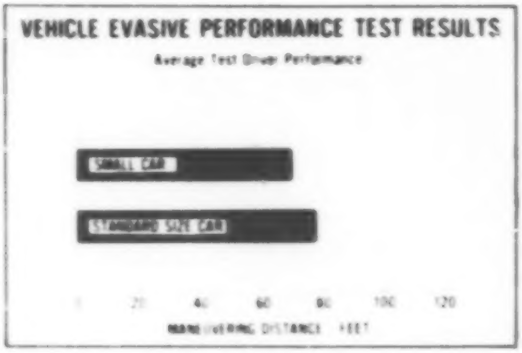


Figure 10

It has been argued that the "open-loop" technique is required to insure reproducibility of test results. To test this hypothesis, General Motors contracted with an independent laboratory to conduct a series of task performance tests at two sites, using two groups of drivers.

The constant radius-turn test and a double-lane-change test were used. Preliminary results indicate that test reproducibility with different expert drivers is definitely possible, while differences in test areas can have a significant influence.

The second objection to the use of the expert driver has been that his performance does not reflect the performance of the typical driver (Figure 11). We have studied the influence of driver skill on task performance test results and found that driver skill varies appreciably. The expert driver had a short learning period and could quickly find and maintain a performance indicative of the vehicle's limits. This of course is important for repeatability in the data.

But perhaps the most interesting result is in connection with our efforts in the field of driver education. Of all the more or less typical drivers we have trained to take advantage of the capabilities inherent in current passenger cars, the reactions have invariably been that: first, vehicle handling qualities under extreme maneuvering conditions are better than they thought and secondly, their own individual driving skills could be improved.

For some time, General Motors' Program for Advanced Driver Education has been made available for traffic safety centers throughout the United States. A two-year study within the Sheriff's Department in Oakland County has produced some thought-provoking results.

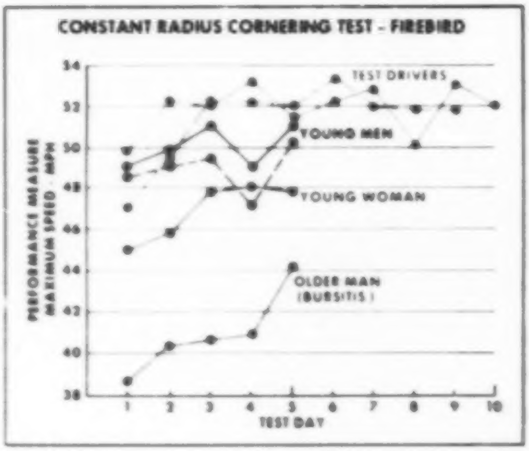


Figure 11

OAKLAND COUNTY SHERIFF'S DEPARTMENT STUDY

	TRAINED GROUP	UNTRAINED GROUP
NO. OF ACCIDENTS	5	10
INJURIES	0	2
LOST DAYS	0	87
LOST WAGES	0	$ 3500.00
VEHICLE DAMAGE COSTS	$1446.50	$11247.10
VEHICLES TOTALED	0	3
TOTAL COST	$1446.50	$14747.10
AVG. COST/ACCIDENT	$ 289.00	$ 1474.00

Figure 12

In a group of 40 drivers, 20 were given the advanced training, while the other half were not (Figure 12). Vehicles and driving conditions were, of course, kept as near identical as possible between the two groups. The results were that, in the trained group, the number of accidents was halved, injuries and lost wages were reduced to zero and both the total costs and the average cost per accident were reduced sharply.

To summarize, there is evidence that vehicles are improving to meet traffic system demands. There is also evidence of uniformity in handling quality of current production vehicles. Because the role of the vehicle in accident causation lacks definition, it is suggested that improved causation information be obtained to aid in selecting safety-related handling tests and to establish the cost effectiveness of handling standards. It is also suggested that the task-performance approach to handling standards be more thoroughly evaluated and that advanced driver education in maneuvering skills be employed wherever possible.

Dr. Limpert: I would like to comment on what Mr. Sorsche of Daimler-Benz said earlier that I'm very happy that 105a is basically acceptable to Daimler-Benz. I would also like to dispel some of the concern over vehicle handling, as far as rulemaking is concerned, and some of the excessive emphasis in my personal view, that has been given to the work of HSRI, and others, with the open loop work that comes out of NHTSA at the present time. It's only a step in a direction of trying to understand vehicle handling. In my judgment, the term vehicle handling should not be attached to a report that deals only with open loop tests. Generally, vehicle handling is interpreted as the capability of the vehicle and the driver to follow or to successfully carry out a maneuver. If you leave the driver out, in my judgment, you don't follow that definition any more. But, the people generally get a lot of mileage out of the words vehicle handling because we all think something new is coming, but it's still something old. It's important that we understand and study the vehicle dynamics aspects of things. As Mr. Scott and Fran DiLorenzo indicated, we, and other people in the automotive community, are moving now towards Phase II, which includes the driver in the research. We try to find out what the dynamic parameters of the vehicle are, or to which dynamic parameters of the vehicle the driver is sensitive. At the present time I don't even know what they are, where they are, or if we will find any. If there are advance notices of proposed rulemaking forthcoming, I would like for you to write NHTSA and state opinion and experience, especially as far as test procedures are concerned, and provide any help you can give us, because vehicle handling as Mr. Sorsche indicated, is not an area where we know even the direction to follow. We'd like to make the third step after we make the second, and we don't want to miss any step.

One more comment to what Mr. Sorsche said about the braking efficiency on the rear axle. Should it be higher or should it be lower than that on the front? I'm not quite sure how that relates to tires whose peak of sliding performance is not nearly the same. For example, on a slippery surface where the peak performance is much higher than the sliding performance, the change in deceleration you would get by locking the front wheels with the rear still rolling may not result in an increased vehicle deceleration rate. Some of the data we have indicate that when you lock the front wheels, they reduce in braking capability and you may not get the highest deceleration.

I fully agree with him on dry surfaces or those surfaces where for the tire/road interface the peak value and the sliding value are nearly the same or constant.

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

The following report (printed verbatim) was submitted for inclusion in the seminar minutes by Alfa Romeo.

FILIPPO SURACE
Alfa Romeo
Italy

STEERING AND HANDLING

In previous conferences we dealt with the following two subjects concerning the problem of road holding qualities:

- Stability and handling of the car on slippery surfaces
- Wheel adherence on uneven surfaces.

Let us summarize our point of view concerning these two subjects:

STABILITY AND HANDLING

We believe that the greatest obstacle to defining a safe car, is the uncertainty of the dynamics required. This uncertainty is mostly due to the fact that the dynamic behavior of the car is considerably influenced by the behavior of the driver which, in our opinion, is not sufficiently known.

In order to overcome these difficulties we initiated, some time ago, a theoretical and practical research program on the driver-car association to determine the essential reactions of the driver and, hence, the limit of acceptability of car dynamic characteristics. In particular, this theoretical and practical research follows the phases indicated below:

- Experimental study of driver-car association.
- Interpretation and generalization of the experiment through sufficiently reliable mathematical application. It is, in fact, practically impossible to carry out completely satisfactory experiments that would allow the clear understanding of the part each of the important dynamic factors plays in the movement of a car. This is mostly due to the driving characteristics which vary from driver to driver, and, for each driver, at different time periods. It is, however, possible to accumulate sufficient experimental data on the average behavior of the driver and to use them to estimate, through an appropriate mathematical formula, the influence of those sectors that concern the movement of the car to verify:
- Calculation of the limit of acceptability of the dynamic characteristics of a car.
- Examination of the objective experiments in order to determine if the dynamic behavior of the car remains within the above limits.

During the present conference, we will explain the latest progress of our program to find a mathematical formula simulating the driver-car association. We will explain the positive and negative aspects of our program.

With regard, however, to the quantitative aspect, i.e. the limit of acceptability of the dynamic characteristics of a car, our present experience advises against expressing a firm opinion. We believe that the problem has not yet been studied sufficiently in depth to allow a quantitative definition of these limits. We hope that whoever may have undertaken similar studies, is of the same prudent opinion.

ADHERENCE ON UNEVEN SURFACES

Despite the progress made in the construction and maintenance of roads, this problem, unfortunately will remain important for some time to come; its importance to safety is obvious.

The road adherence of the wheel must be considered, be it in cases where the car meets with an isolated obstacle or be it on roads with a number of such obstacles.

ESV has taken into account the first aspect of the problem but not the second one. Therefore, we have begun studies of an uneven steering pad on which to carry out tests to determine the adherence limits of front and back wheels. Experience gained with racing cars has been useful in this context.

We have in particular tried to overcome the difficulties encountered in the construction of a steering pad with continuous undulations. We, therefore, submitted to the Washington conference our experiments with steering pads with unevenly distributed obstacles with power spectrums very similar to those found on the roads.

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During the present conference we will provide some of the most interesting test results.

A new steering pad is being developed which, based on experience gained, will eliminate some of the inconveniences of the previous constructions.

Subsequently, we constructed some of these steering pads and carried out tests with different car types. The usefulness of these steering pads was confirmed in obtaining optimum adherence conditions.

ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

The following report (printed verbatim) was submitted for inclusion in the seminar minutes by Girling, Ltd.

SAFER BRAKING SYSTEMS

B. INGRAM, *Chief Engineer*
Advanced Control Systems
P. OPPENHEIMER, *Technical Legislation Manager*
Girling, Limited, United Kingdom

INTRODUCTION

The introduction of many new international braking regulations and safety standards has created a worldwide interest in the development of safer braking systems.

This presentation reviews the increasing legislative requirements against the practical experience of the motor industry and the theoretical investigations of the academicians, so that the practising automobile engineer may appreciate the current state-of-the-art of braking systems and the progress towards improved road safety.

SAFETY

The adoption of safer braking systems for vehicles would enable many accidents to be avoided.

This basic objective represents a considerable challenge to the motor industry in which the mechanical engineer can make a real contribution to society.

Safer braking systems can be engineered in several ways:

(1) By reducing the incidence and the effects of brake failures.

(2) By improving the stopping performance of the vehicle.

(3) By reviewing the dymanics of vehicle braking.

(4) By introducing anti-locking brake systems.

Each of these approaches offers considerable scope for the development of new products and systems and each features in the current or proposed regulations and safety standards.

However, every component, assembly or system that is added to the basic braking system for increased safety, will at the same time add to the chance of developing a failure. This fundamental conflict between complexity and reliability is aggravated by the use of deteriorating components, such as fluids, seals, hoses and pipes which have only a limited life; and this life can vary significantly with different operating conditions.

The more such components are used in the braking system, the greater is the probability of a failure occuring. This is illustrated in Figure 1, where the inherently reliable single braking system is represented by a master cylinder with 2 seals; the modern divided braking system is represented by a tandem master cylinder with 4 seals and the next generation of braking systems is represented by a triple master cylinder with 6 seals. Although the chosen scale values may be debatable, the hyperbolic relation is typical and the principle remains, that any increase in system complexity is accompanied by an increased failure probability.

This controversy becomes even more significant when more sophisticated braking systems are under consideration, such as power-hydraulic and anti-locking systems, when the choice between complexity and reliability becomes vitally important.

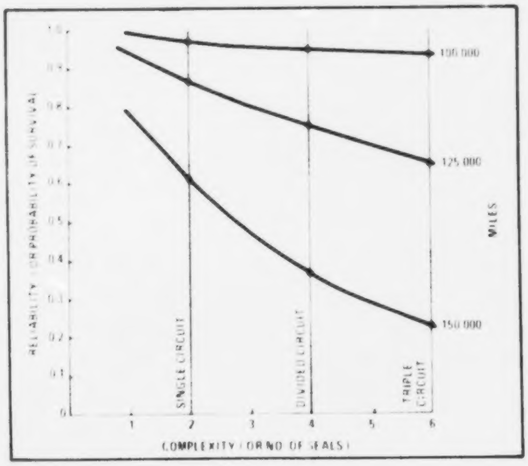

Figure 1

Another factor of increasing importance in this context is the extent and quality of the maintenance and repairs, when garage costs are escalating and service effectiveness is doubtful. The more complex system is likely to require more servicing; on the other hand, any faulty maintenance is not so likely to prove disastrous. In effect, the complex system might be a palliative to poor maintenance.

BRAKING REGULATIONS

Most countries prescribe certain minimum legal performance requirements for the service, parking and emergency brake systems (the emergency performance must be provided after a failure in the service brake system). Unfortunately, the requirement levels vary from country to country; furthermore, many countries demand additional system requirements, warning indicators, component specifications and special environmental tests, thus contributing to a bewildering variety of vehicle specifications. Nevertheless, there are three braking regulations which dominate the international scene and which contain nearly all the most stringent requirements and which are likely to become more widely adopted. The passenger car sections of these three regulations (E.E.C., Sweden and U.S.A.) are summarized in Table 1.

United States Of America

The special emphasis on vehicle safety originated in the U.S.A. after the publication of Ralph Nader's book *Unsafe at any Speed* in 1965. Subsequently, the first Federal Motor Vehicle Safety Standards were introduced in 1968.

The original braking Standards were based on the corresponding S.A.E. (Society of Automotive Engineers) test procedures and performance standards. This aspect of the S.A.E. activities, incidentally, is notably lacking within the United Kingdom engineering institutions and is not adequately covered by the S.M.M.T. (Society of Motor Manufacturers and Traders) or the B.S.I. (British Standards Institution). All three Federal braking Standards, covering hydraulic brake systems, brake hoses and brake fluids, are currently being upgraded; some of the new proposals are shown in the last column of Table 1. There are also many State Laws relating to brake systems and components, such as letter edge-marking of brake linings and the inclusion of re-grinding dimensions on discs and drums.

A special feature of the Federal Standards is their method of approval. In most other countries, every new vehicle brake system must be 'type-approved' before it can be sold to the public; this type-approval is carried out by the national vehicle testing station on a new vehicle submitted by the manufacturer or local distributor. In the U.S.A., however, the vehicle manufacturer must certify that his vehicles comply with all relevant Standards. The Federal authorities may, at any subsequent time, select a new vehicle from a showroom and carry out the appropriate compliance tests. If the vehicle fails any particular test, modifications may have to be introduced before further vehicles are allowed to be sold; these modifications may have to be applied to all vehicles in service and a fine imposed for each defective vehicle sold. This self-certification procedure is therefore particularly onerous on the vehicle and safety equipment manufacturers, bearing in mind the statistical variations in performance that could occur in such a random vehicle selection method. Another interesting feature of the Federal Standards is the Consumer Information; each dealer showroom must provide literature to show the 60 mile/h stopping distance that every individual vehicle can meet.

Finally, the ESV (Experimental Safety Vehicle) program was initiated by the Department of Transportation in 1967 to obtain a 'quantum jump' in vehicle safety by designing new vehicles from fundamental principles, with safety as the principal design parameter, and at the invitation of N.A.T.O., taken up by Germany, Japan, Italy, U.K., France and Sweden. While the major efforts are being devoted to crash-worthiness and passive safety features, resulting in the development of mobile battering rams and bumper cars, all the ESVs will also embody improved braking systems incorporating anti-locking devices. Any proven ESV features are likely to become legislative requirements, probably around 1978, for all new vehicles in the U.S.A.

At the time of Nader's book in 1965, the braking systems on the large and powerful American cars were inferior to their European counterparts. By 1974, however, the Federal braking Standards will represent the most stringent braking requirements in the world.

Sweden

The Swedish Government has taken a particularly strong interest in road safety and issued an exemplary Safety Standard F18–1969, which became effective for 1972 models. F18 contains many novel features, notably the demand for automatic lining wear adjustment by service brake operation and the requirement for brake apportioning to prevent premature rear-wheel lock. The final paragraph in F18, however, has created the greatest technical interest; effectively it states that 4-wheel anti-locking brakes might be

Table 1
INTERNATIONAL BRAKING REGULATIONS FOR PASSENGER CARS

1.0	Territory	E.E.C.	Sweden	U.S.A. (1)	U.S.A. (2)
2.0	Introduction date	≥ 1973	1972	Current	Proposal
3.0	SYSTEM REQUIREMENTS				
3.1	Divided circuits	* yes (from 1.10.74)	yes	yes	yes
3.2	Separate fluid reservoirs	—	—	—	yes
3.3	Automatic adjustment by service brake	—	yes (or 4.3)*	—	—
3.4	System strength	—	yes	—	—
3.5	Activation time	yes with power assist	yes	—	—
3.6	Pedal travel limits	yes	yes	—	—
3.7	Vacuum reservoir	—	yes	—	—
3.8	No asymmetric braking	—	—	—	-
3.9	Apportioning	—	—	—	-
3.10	No redundant members	yes	yes	—	—
4.0	WARNING INDICATORS				
4.1	Pressure loss	yes	yes (or 4.2)	yes (or 4.2)	yes
4.2	Fluid level	yes (or 5.5)	yes (or 4.1)	yes (or 4.1)	yes
4.3	Lining wear	—	yes (or 3.3)	—	—
4.4	Vacuum or servo failure	yes	yes	—	—
4.5	Parking brake 'applied'	—	—	—	yes
4.6	Apportioning or anti skid failure	—	—	—	yes
5.0	COMPONENT SPECIFICATIONS				
5.1	Bundy pipes	—	yes	—	—
5.2	Brake fluid	—	yes	yes	yes
5.3	Hydraulic brake hoses	—	—	yes	yes
5.4	Air or vacuum hoses	—	yes	—	yes
5.5	Transparent reservoir	yes (or 4.2)	yes (or 4.2)	—	—
5.6	Lining material approval	yes	—	yes	yes
5.7	Lining inspection holes	—	—	—	yes
5.8	Disc drum machining marking	—	—	yes	yes
6.0	TEST CONDITIONS				
6.1	Road surface	good adhesion	0.8 μ	dry concrete	skid number 75
6.2	Test weight	driver only and full load	any load	full load	light load and full load
6.3	Test speed	80 km/h	80 km/h	60 mile/h*	60 mile/h*
6.4	Stability requirements	no locking and no deviation	no deviation and special locking conditions	no deviation from 12 ft lane	no deviation from 12 ft lane
7.0	SERVICE BRAKE				
7.1	Stopping distance	$V/10 + V^2/150$	—	—	185 ft
7.2	(Equiv.) deceleration	59% g	59% g	62% g *	(65% g mean) *
7.3	Pedal effort	50 kg	50 kg	120 lb	120 lb
8.0	EMERGENCY BRAKE				
8.1	Stopping distance	$V/10 + V^2/75$	—	646 ft	388 ft
8.2	(Equiv.) deceleration	29.5% g	29.5% g	(18.5% g mean)	(31% g mean)
8.3	Pedal effort	50 kg or	50 kg or	200 lb	150 lb
8.4	Handbrake effort	40 kg	40 kg	—	—
9.0	PARKING BRAKE	FL only	(on 0.6 μ)		(on dry concrete)
9.1	Gradient hold	18%	16%	30%	30%
9.2	Handbrake effort	40 kg or	40 kg or	—	90 lb or
9.3	Footbrake effort	50 kg	50 kg	—	125 lb
9.4	Dynamic test	in motion	20 km/h	—	—
9.5	Deceleration	—	—	—	—
10.0	OTHER PERFORMANCE				
10.1	Servo-failed	yes	yes	yes	yes
10.2	'In-service'	—	−10%	—	—
10.3	Fade test	yes	yes	yes	yes
10.4	Water test	—	yes	yes	yes
10.5	Other tests	various speeds and 'in gear' tests	high speed test	high speed and light load tests	high speed and spike stops
11.0	REMARKS	*Each divided circuit must provide min. 18% g (FL) and 15% g (DO) for 70 kg pedal effort	*Postponed to 1.1.73/74	*Also 30 mile/h (100 lb) and 80 mile/h (150 lb)	*Also 30 mile/h (100 lb) and 80 mile/h (150 lb)
FOR COMPARISON ONLY Stopping distance From 60 mile/h	Service Emergency	234 ft 436 ft	234 ft 436 ft	226 ft 646 ft	185 ft 388 ft

prescribed for around 1975. A subsequent meeting of experts in Stockholm doubted the practicability of enforced legislation by 1975.

E.E.C.

The E.E.C. (European Economic Community) braking regulations are contained in a Directive, which was approved by the Brussels Parliament in July 1971. These regulations are mandatory within the Common Market area and must be adopted into the various national legislatures within 18 months. They could become applicable in the U.K. during 1973.

In technical terms, the braking performance requirements are not very onerous and are generally based on the E.C.E. (Economic Council for Europe) Regulation 13. This regulation is notable for its detailed classification of vehicle categories and the associated braking performance requirements in terms of stopping distances.

REDUCING THE INCIDENCE AND THE EFFECTS OF BRAKE FAILURES

Fortunately, the number of brake failures is relatively small; nevertheless, their effect is often catastrophic. To improve this situation requires a better understanding of the failures that occur in service and particularly of those faults that cause accidents. In the course of such investigations, some distinction might be made between:

(1) Hydraulic failures, such as seal leakage, burst hose or pipe fracture; which usually occur suddenly and without warning, often under emergency conditions when extra pedal effort is applied.

(2) Mechanical failures, such as corrosion, seizure or wear; which normally occur gradually and may be avoided by proper maintenance.

(3) Stability failures, such as 'pulling' or wheel locking, which are most prevalent with poor road surface conditions, but are often due to (2) or basic deficiences in the brake distribution compromise.

Guarantee claims, spare parts service sales analyses and manufacturers' special field investigations are all used to elicit information on failure statistics.

Nevertheless, major sources of such information are:

Periodic Inspections

The annual M.O.T. (Ministry of Transport) test is typical of this category, which could provide much useful information on which to base design improvements; but the present results are insufficiently detailed. The survey by the Swedish Government Testing Stations shows what can be done during a more thorough examination; 25 per cent of all cars failed on brakes, and this summary is further broken down to reveal that half of these failures were due to brake pipe corrosion. Appropriate action by the Swedish legislators has subsequently specified an improved corrosion resistance for all new cars.

While this type of examination can expose the weak links in braking systems, it does not necessarily reveal the brake failures which cause accidents. For example, an ineffective parking brake would fail the M.O.T. test, but is unlikely to cause an accident. Conversely, skidding on wet roads is claimed to cause 25 per cent of all accidents, but no periodic test would be expected to show this feature, which could be fundamental to the orignal type-approved installation.

Accident Investigations

It is generally agreed that at least 90 per cent of all accidents are caused by human errors; but this does not mean that these accidents might not have been avoided by more efficient braking systems. In fact, other statistics show that braking is involved in more than half of all accidents and this provides a better appreciation of the potential savings bearing in mind that 150,000 people are killed annually on the world's roads.

The results of typical accident studies by the Transport and Road Research Laboratory and the police are often difficult to analyse. The total number of such accidents are fortunately relatively small in the U.K. and doubtful to form the basis of reliable statistical evidence. Nevertheless, there is no doubt that skidding and loss of control, often as a result of rear-wheel lock, is a major source of accidents and apparently contributes to 75 per cent of all accidents in icy conditions. Similarly, statistics have been presented to show that 25 per cent of all braking accidents could have been avoided by using divided hydraulic circuits.

Further investigations along these lines are proposed under the U.K. action for the ESV program, and although the difficulties are enormous, ultimately only this type of work can demonstrate the effectiveness of the efforts of the legislators and the manufacturers towards improvements in vehicle safety.

Divided Circuits

The obvious method of reducing the effects of hydraulic failures is to divide the hydraulic 'transmission' into two or more separate circuits. Any single hydraulic failure will then enable a certain residual emergency performance to be achieved with the remaining intact footbrake circuit. Accordingly, divided circuits are becoming an increasingly popular legal requirement and five different dual-circuit systems have been adopted by various vehicle manufacturers (Figure 2).

The choice between these solutions is of considerable importance in terms of cost-effectiveness. In practice, cost is often of overriding importance. System *A* will always be the cheapest and simplest solution and is by far the most popular. The subsequent order of merit for this parameter is *B*, *C*, *D* and *E*. System *A* provides the only satisfactory solution for vaporization which is a serious temporary brake failure experienced in hydrostatic brake systems.

There are several other parameters which influence the choice of dual-circuits, but the most important factor is the potential deceleration of each individual circuit and the associated vehicle stability. Figure 3 enables this vital safety parameter to be assessed for each of the five dual-circuit solutions.

This example is for a conventional 2-litre saloon with a front-to-rear fixed braking ratio of 70:30, chosen to meet the Swedish legal wheel-locking requirements. On a 0·8 adhesion surface and with any vehicle loading, no wheels may lock before 0·6 g deceleration and the rear wheels must not lock before the front wheels above 0·6 g.

The rear circuit of System *A* can only just meet the 0·3 g legal emergency brake requirement and is accompanied by rear-wheel locking; it is undoubtedly the worst solution in terms of road safety. Surprisingly, System *B* also provides only a marginal residual performance, albeit with the front wheel locking first and thus retaining stability; and even steering control may not be entirely lost, because the other front wheel remains rolling. Systems *C* and *D* provide perfectly adequate emergency performance, while System *E* can theoretically achieve the full service brake performance from each individual circuit.

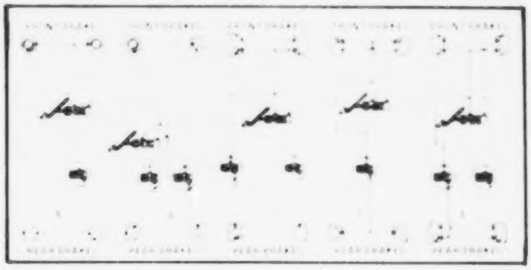

Figure 2

Front-wheel-drive cars suffer from a very low rear-axle braking potential; at the same time, the need for good steering characteristics demands a minimum steering ground offset and for these two reasons, System *B* offers distinct advantages.

A side-effect of the safety legislation for divided circuits is the associated free pedal travel which may be as much as 3 inches in the failed condition of the current hydrostatic tandem master cylinder. If the circuit failure occurs during a brake application, then there will be an accompanying delay period during which the extra pedal travel is executed and the hydraulic pressure is reapplied in the surviving circuit. The driver's reaction to this combination of extra pedal travel and delay period, plus reduced deceleration with increased stopping distance, plus the likelihood of locked wheels, instability and loss of control is likely to be disastrous. Particularly in view of the general publicity and advertising claims about 'duplicated' and 'fail-safe' braking systems, which might suggest to the consumer that a brake failure will not materially affect his braking potential.

Warning Indicators

Several types of such devices are becoming legal requirements.

Brake Fluid Level Indicators – These indicators constitute a simple and desirable safety aid. Many countries already prescribe such indicators, although some permit a transparent fluid reservoir as an alternative. Several proprietary indicators, usually based on an electrical contact on a floating member, are available. An alternative solution, which avoids

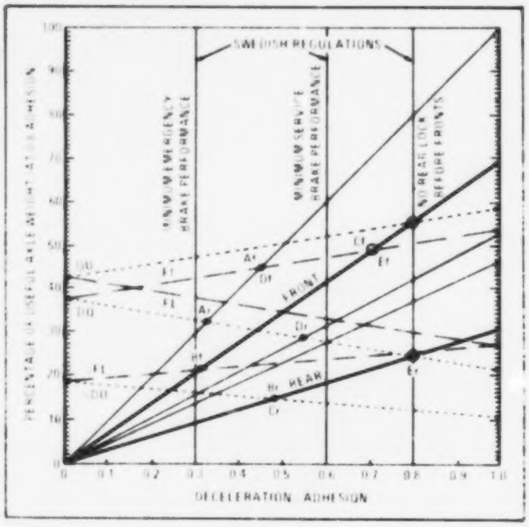

Figure 3

the probe contact process, uses a floating magnet to operate a reed switch. Another solution, which avoids floats, contacts and switches, continuously monitors the electrical resistance between two probes at the minimum fluid level mark and detects the change in resistivity when the gap contains air instead of brake fluid.

Brake Failure Indicators These are an essential part of any divided hydraulic system. Most regulations permit a fluid level indicator as a circuit failure warning device; such an indicator may detect a serious leakage failure *before* a brake application a vital safety feature, distinctly lacking from the conventional pressure-differential switch, which only operates *during* a brake application.

Lining Wear Indicctors These have been fitted on several disc brake applications for many years. Usually an electric wire is embedded in the lining material to make contact with the disc when pad replacement is due.

An unsatisfactory feature of all the current warning indicators is that they only check the bulb circuit; a broken wire or loose contact will prevent the indication of a brake failure and the driver may well assume the opposite, since the bulb circuit continues to function correctly.

One solution has been the development of an electrical circuit which operates on the 'break' principle, whereby any system failure (e.g. low fluid level) or electrical failure (e.g. loose wire) will cause the red warning light on the dashboard to illuminate. Such fail-safe devices can be applied to any warning indicators and to the more complex power-hydraulic and anti-locking electrical circuits, where failure indication is becoming increasingly more difficult. Incidentally, a flashing red light might seem more appropriate to indicate a brake failure rather than a continuous red light, which could easily be mistaken for low oil pressure or low generator charging.

IMPROVING THE STOPPING PERFORMANCE

Being able to bring a vehicle to a standstill is obviously a major factor in avoiding accidents. At the same time, one may also wish to retain steering control and avoid skidding during maximum braking, and under all operating conditions.

One possible solution would be to increase the brake torque capability, but unfortunately, the available tire-to-road adhesion precludes declerations over 1 g for road vehicles, although racing cars with special tires can achieve more than 1.5 g. Such rubber compounds for road vehicle tires might provide an attractive safety feature. The use of aerofoil wings, which increases the load on the wheels at high speeds, also enables additional deceleration to be achieved. Other novel forms of braking may provide higher decelerations in the future: for example, suckers (reverse of the hovercraft principle, already used in motor racing for stability during cornering), magnets, retro-rockets and anchors.

In the meantime, the maximum deceleration remains limited to about 1 g and most regulations specify a retardation of 0.6 g on a reasonably good, dry surface. The precise test surface definition, which varies from country to country, can constitute a significant difference in the achievement level.

Another basic factor liable to different interpretations is the definition of 'deceleration'. Figure 4 illustrates a typical vehicle deceleration process, from the time when the signal to stop the vehicle is given (used in the French regulations) until the vehicle has stopped. The maximum deceleration is the most popular value, equivalent to the 'fully developed retardation' (Sweden), 'brake effort' (Germany) and the effectiveness or efficiency measured on roller dynamometers. During vehicle tests, the steady U-tube reading would be representative of the 'maximum' deceleration.

The 'mean' deceleration is based on the stopping-distance, usually from the time when the brake pedal is applied; it cannot easily be read off during the stop and is therefore not popular when a certain deceleration must be achieved or maintained. On the other hand, in terms of road safety and consumer information, stopping-distance is the correct parameter.

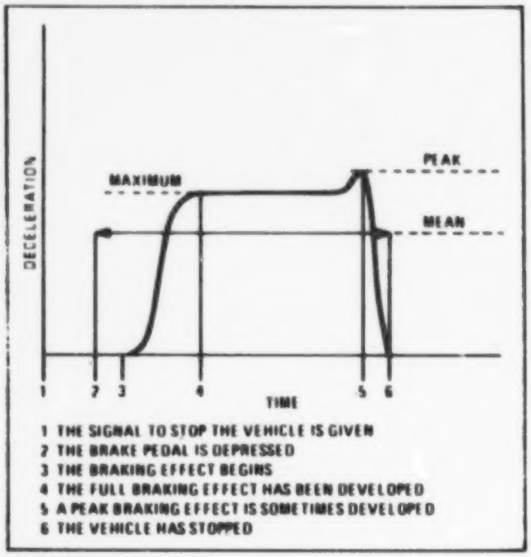

Figure 4

The importance of this terminology problem may be appreciated from practical tests, which have shown the maximum deceleration to be between 20 and 25 per cent above the mean level during the same stop. Finally, there is the 'peak' deceleration, which sometimes occurs at the end of the stop and can be recorded with pendulum instruments; this reading is admissible in certain countries, including the U.K.

The associated legal maximum pedal effort for 0.6g is usually 50 kg. This represents an excessive demand during panic braking, especially for women drivers. A British design aim used to be 1 lb per 1 per cent g and a realistic legislative target might be around 35 kg for 0.6 g which corresponds to the current Australian requirements.

Brake linings play an important part in road safety. Their effects can increase stopping distances and/or cause premature wheel locking. Accordingly, the legislators have adopted several additional test schedules to check the suitability of brake linings under more arduous conditions. The Swedish type-approval, for example, includes a fade test, water recovery test and high-speed test. In the case of E.C.E. and E.E.C., the type-approval document specifies the lining material which was tested. No alternative may be fitted during the life of the vehicle; this appears unduly restrictive.

The U.S.A. approach is generally more acceptable, whereby all approved brake linings are classified according to their hot and cold friction levels, i.e. FF, FG. Lining replacement is then permitted with new linings of the same friction classification. The actual classification is made on a Chase-Link machine, the results of which do not correlate with the lining performance on the vehicle. Attempts are being made in the U.K. to establish a more acceptable solution and an appropriate credibility study is in progress.

Quality control of linings can also make an important contribution to road safety, because the friction level of a lining grade can vary by as much as 25 per cent during production from batch to batch.

Finally, the brake assemblies themselves have a considerable influence on the stopping performance and there may be a tendency to forget the progress that has been made since the early days of four-wheel braking with cable-operated mechanical drum brakes.

DYNAMICS OF VEHICLE BRAKING

The vehicle stopping performance can be improved by modulating the braking forces with apportioning or anti-locking systems.

Such devices help to increase the utilization of the available tire-to-road adhesion, either by modifying the braking distribution in line with the dynamic weight distribution or by controlling the braking forces to operate near optimum slip conditions.

Elementary vehicle dynamics reveal that acceleration or deceleration of any vehicle will vary the weight distribution on the wheels; this also applies transversely across the vehicle during cornering. During straight-line braking, weight is transferred from the rear axle to the front axle and it follows that the ideal braking ratio for any particular vehicle should take account of:

(i) The vehicle weight.
(ii) The static weight distribution.
(iii) The height of the center of gravity.
(iv) The deceleration.

A fixed-ratio braking system only provides optimum performance at one particular adhesion level for any one loading condition. In practice, a compromise between dry and wet roads tends to achieve optimum performance around 0.7 adhesion. At any other adhesion level, either the front wheels or the rear wheels will lock before the optimum deceleration can be achieved. Some form of variable-ratio braking system is therefore desirable to approach the ideal requirements and at the same time eliminate the dangerous premature rear-wheel locking situation.

Proportioning Valves

The simplest method of modifying the conventional fixed-ratio braking distribution is to add a pressure modulating valve in the rear brake line. Three basic types of such valves are available and each can be manufactured with pressure-limiting or pressure-reducing characteristics:

(1) Deceleration conscious valves.
(2) Pressure conscious valves.
(3) Load conscious valves.

The relevant advantages have to be balanced between the performance achievement level and the cost-simplicity of design and installation on any particular vehicle.

The load conscious pressure-reducer provides the best performance but is the most expensive and the linkage is vulnerable to damage and maladjustment. In practice, lining friction coefficients may become extremely critical; a car fitted with a variable-ratio system matched to prevent rear-wheel locking throughout the adhesion range could be very adversely affected by a slight increase in rear brake lining effectiveness. Thus the potentially safer system could then exhibit a greater degree of instability.

Adhesion Utilization

The principle of adhesion utilization, its mathematical derivation and its significance in assessing brake system performance have been described by Rouse [1]*.

Figure 5 shows the improved adhesion utilization (and the absence of rear-wheel locking) when an apportioning valve is added; in fact, it is often possible to achieve in excess of 80 per cent utilization over a very wide range of adhesion conditions from dry concrete to wet ice.

Unfortunately, the great limitation of all such apportioning devices is that the brake system itself cannot prevent the driver ultimately exceeding the maximum safe deceleration and then locking the wheels.

*References are given in the Appendix.

The Significance of Locked Wheels

Three possibilities can arise: either the front wheels, the rear wheels or all the wheels may lock. If only the front wheels lock, or if all wheels lock at the same instant, the vehicle continues in a straight line. If only the rear wheels lock, the vehicle tends to yaw. If all four wheels do not lock at the same instant, the behavior of the vehicle is dependent on the time interval between the locking of the rear and front wheels. For example, half a second can be sufficient to turn the vehicle.

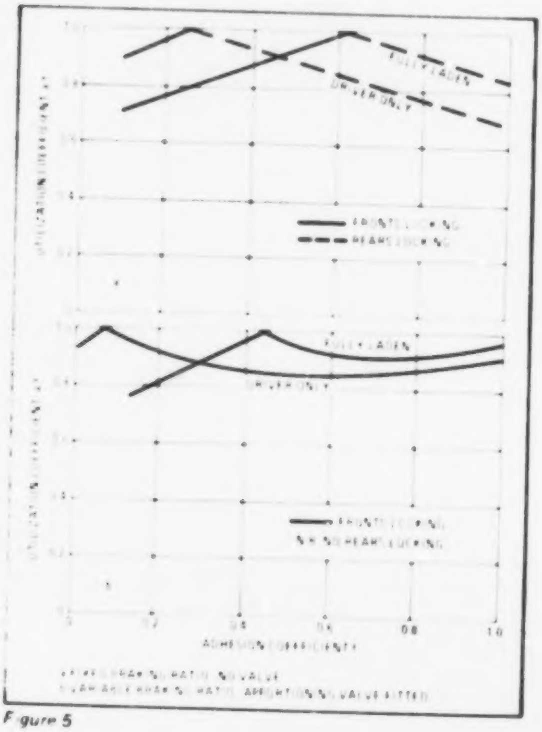

Figure 5

The variable-ratio braking system performs well on uniform adhesion surfaces, but cannot adapt adequately when the vehicle is straddling two different surfaces. Under such differential-friction conditions, if one or both rear wheels lock and irrespective of whether the front wheels are locked or not then the vehicle will develop an exceptional degree of yaw.

Summarizing, locked wheels affect two specific handling characteristics: stability and controllability. Stability can exist without controllability; in contrast, controllability implies stability but enables the driver to create instability.

Rear-wheel Control

Any system which protects the rear wheels from locking during emergency braking will almost entirely eliminate angular deviations and thereby retain stability even though steering control may be impaired or lost completely by locking the front wheels.

Figure 6 indicates the reduction in yaw angles with maximum braking on two sets of differential-friction surfaces. Three conditions of the same car were evaluated.

(1) The standard braking system.
(2) The rear brakes disconnected.
(3) The rear brakes fitted with an anti-locking system.

Both conditions (2) and (3) eliminate the rapidly increasing yaw angles of (1) but (3) also reduces the stopping distances significantly. The most descriptive terminology for such rear-wheel anti-locking systems would be 'anti-yaw' systems.

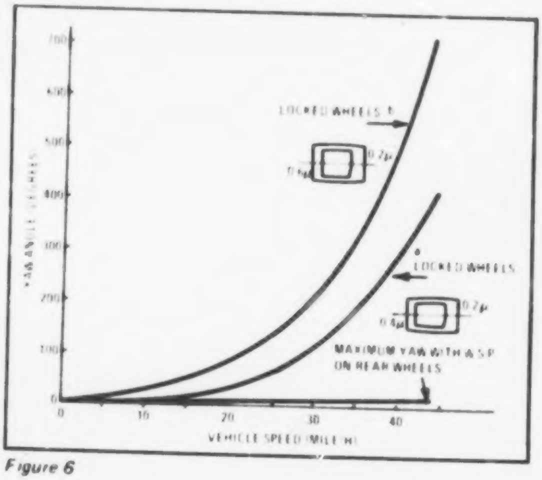

Figure 6

Much has been written about rear-wheel control systems, often aimed at discrediting their contribution to road safety. Yet the ability to maintain lateral stability under any straight-line braking conditions, without loss in stopping distance, must represent a very significant safety feature especially where a large proportion of accidents and fatalities occur on multi-lane highways. Another vital feature of anti-yaw systems, compared with any other anti-locking system, is that if the car is completely unstable, applying the brakes *hard* will help to restore stability.

Four-wheel Control

Locked front wheels cause a loss of steering control; hence it would be desirable to protect them from locking during braking. However, if the rear wheels are still allowed to lock, then the vehicle will deviate and if it exceeds an angle of about 20°, directional control may be lost completely even if all brakes are subsequently released. It is therefore essential to control all four wheels of the vehicle.

The basic combinations for controlling the four wheels are usually confined to three arrangements (Figure 7), which can theoretically provide similar stopping distances during straight-line braking on uniform adhesion surfaces. On differential-friction surfaces or when the combined requirements of steering and braking are taken into account, axle control becomes less effective than wheel control, because in controlling an axle there is a choice of:

(i) Sensing the slowest decelerating wheel (selecting HIGH).

(ii) Sensing the mean deceleration of the wheels (selecting AVERAGE).

(iii) Sensing the fastest decelerating wheel (selecting LOW).

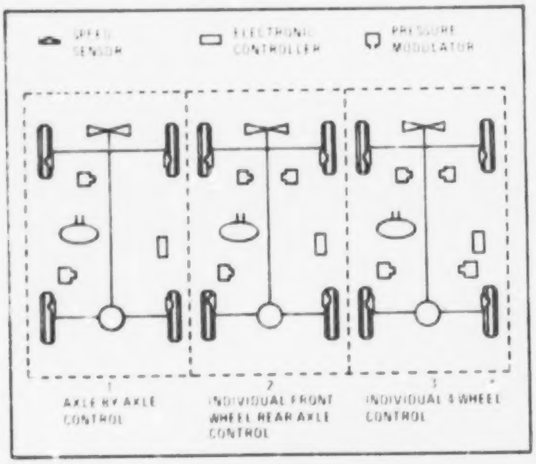

Figure 7

If controllability is the prerequisite, then (iii) is the best solution; but this may tend to increase the stopping distances, especially under differential loading conditions.

Reverting to the choice of arrangements (Figure 7), the simplest and cheapest arrangement (1) is deficient in achievement terms, particularly during combined steering and braking. Systems based on (2) provide a sensible compromise between performance and complication, but systems based on (3) appear to offer the best performance. However, compliance in suspensions affects the steering and handling characteristics, and differential torques on some divided rear axles can induce rear-wheel steering, which will vary in intensity from side-to-side of the axle because of the cyclic nature of the braking torques. This may seem pedantic, but the key to vehicle stability is good rear-wheel control smooth, progressive and at rates of slip generally less than at the front wheels. This is further explained in reference [2], where the description of combined steering and braking on a constant radius path illustrates some of the problems facing the designer of a four-wheel anti-locking system (Figure 8). For example, it seems unreasonable to achieve a shorter stopping distance under one condition, at the expense of aggravating the transitional stability during some other maneuvers. Safety is absolute and should not be compromised.

ANTI-LOCKING BRAKE SYSTEMS

So far, only locked and rotating wheels have been considered, but this is insufficient. Wheels must be maintained at controlled levels of slip, so that they can make the best use of the available adhesion—consistent with the retention of steering control. For the purpose of this presentation, such a technique will be described as W.S.P. (wheel slide protection).

The adhesion characteristics of different tire-to-road combinations vary considerably, but can be

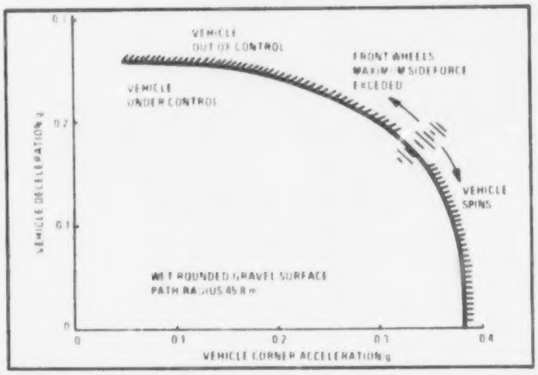

Figure 8

identified by the 'slip' curves such as Figure 9, which shows typical braking force coefficients and sideways force coefficients; the latter are especially relevant during combined steering and braking, showing that locked wheels cause a complete loss of lateral stability.

The W.S.P. Process

The driver's demand pressure generates the brake torque and the relative velocity between the wheel and road changes as the tire slip increases to develop the necessary grip. During this period, the vehicle will experience a proportionate increase in deceleration. If the brake torque continues to increase, then the limit of adhesion will be reached and the brake becomes 'over-pressurized'. If the brake-line pressure is not relieved immediately, the wheels will lock within milliseconds because the excess torque is only used to brake the wheels, which are about 1/40th of the total mass moment of inertia of the vehicle. The same degree of over-pressurization will result in the same order of wheel deceleration, irrespective of whether the vehicle is on ice or dry tarmac. The main effect of the difference in the adhesion level lies in the rate at which the wheel will reaccelerate. If the brake-line pressure is relieved by the same amount below the adhesion limit, then the same wheel would reaccelerate about ten times faster on the tarmac. Hence, good anticipation is a prime requirement of any W.S.P. system and few would challenge the superiority of electronics when speed of decision is so vital.

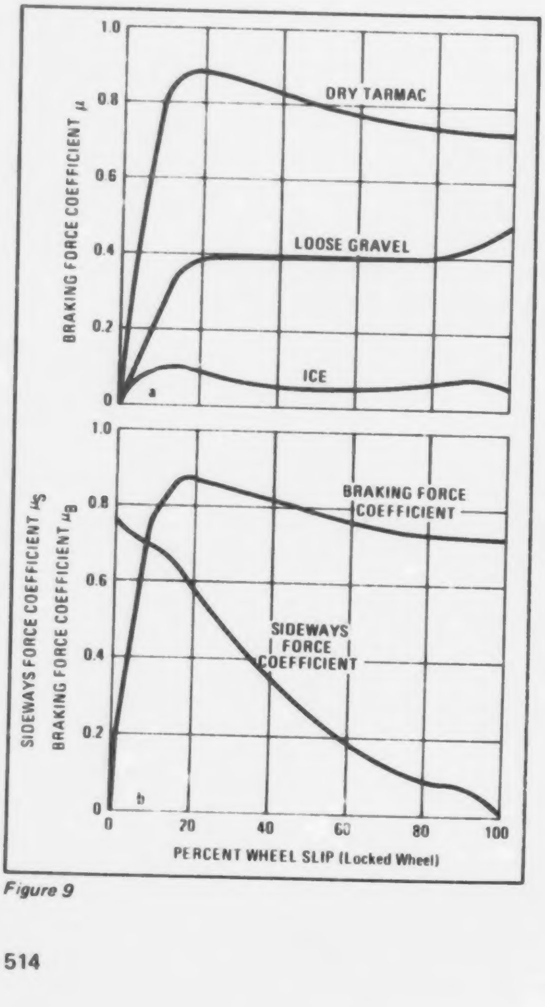

Figure 9

W.S.P. Techniques

The basic parts of a W.S.P. control loop comprise:

(1) A transducer, which may measure speed, pressure, torque or a combination of these parameters to provide information on the behavior of the wheel.

(2) A logic controller, which may be an electronic digital or analogue device, capable of analyzing the signals from the transducer and applying them to command the actuator.

(3) An actuator, which responds to the controller's commands and modulates the pressure in the brake line.

How does such a control loop enable the braked wheel to make maximum use of the available adhesion? Results show [3] that the braking force coefficient may reach a peak anywhere between 7 and 25 per cent slip, although loose gravel and fresh snow are notable exceptions. Furthermore, it is difficult to measure the vehicle speed accurately during the braking process without an independent means, such as a 5th wheel or radar. For these reasons, it is not currently realistic to rely on accurate and continuous measurements of slip to control the wheels during the braking process. Hence, all commercially developed W.S.P. systems operate on the 'on-off' principle, whereby successive approximations by way of controlled wheel velocity excursions are made in an attempt to achieve maximum use of the available adhesion. These wheel velocity trajectories should ideally be small and close to the peak of the slip curve, in order to achieve the maximum benefit of the braking force coefficient.

Failure Modes

By definition, any safety system should never make the vehicle less safe than with standard brakes; stability failures must be considered as well as system failures.

Consequently, W.S.P. systems should guarantee the emergency footbrake performance in the event of any electrical, mechanical or hydraulic failure, and without recourse to automatic devices whose satisfactory function is not known or which are normally at rest. Any such failure must also be optically indicated to the driver.

Rear-wheel Systems – If a rear-wheel system is working correctly, then the vehicle is always stable. If the rear-wheel system fails to operate, then the braking system should return to standard and provide the normal service brake performance. If the rear-wheel system fails such that pressure cannot be generated at the rear brakes, this becomes equivalent to a half-system failure in a conventional divided braking system and the vehicle remains stable. Neither condition is less safe than a standard braking system.

Four-wheel Systems – Such systems will not induce instability by themselves, but because the driver is able to steer the vehicle during braking, he can make excessive demands on the available adhesion and created instability. Unlike the rear-wheel system, the application of braking makes little contribution towards regaining stability, once control has been lost. Hence, the term 'anti-skid' is a misnomer and creates a wrong impression.

Since four-wheel systems should ideally control each front wheel independently, during a W.S.P. system failure either front wheel could be returned to normal braking or be completely disconnected. A failure of the rear portion of a four-wheel system highlights another popular fallacy: to return the rear brakes to standard and leaving W.S.P. on the front wheels makes the vehicle distinctly unstable. It may be considered desirable to return *all* brakes to standard, but an alternative and possibly safer solution is to allow the rear brakes to fail *unsafe*, i.e. non-functioning, while retaining W.S.P. on the front wheels.

Assessing W.S.P. Systems

Many articles have been written about W.S.P. systems, each claiming bigger and better reductions in stopping distances. Some explanation of these claims seems necessary because some systems appear almost magical in their capability.

The theoretical maximum reduction in stopping distance depends on the difference between the peak and slide braking force coefficients (Fig. 9). If the peak is substantially greater than the slide value, then the W.S.P. system can give an appropriate increase in performance and hence reduce the stopping distance considerably. On the other hand, if there is no clearly defined peak, or the peak occurs at 100 per cent slip (such as on gravel and loose snow), then no W.S.P. system can improve on the locked-wheel stopping distance. It is therefore clearly necessary to know the actual slip curve, in order to judge stopping distance improvements. There are other aspects of W.S.P. systems which have greater relevance than stopping distances; for example, rear-wheel systems can reduce yaw considerably and it is obviously essential to evaluate this capability. Four-wheel systems involve the understanding of vehicle behavior under combined steering and braking. This condition was previously never considered by the brake engineer, because vehicle handling characteristics were usually based on steady-state conditions such as steering pad tests. Test procedures are now being evolved, which specify combined steering and braking on constant radius curves and through chicanes.

Because of the complex nature of the operational characteristics of W.S.P. control loops, vehicle tests alone are insufficient to assess W.S.P. systems in the development stages. For example, some important parameters which cannot readily be monitored during vehicle tests concern the details of the particular surface slip curve and the precise position of each wheel on this curve. Fig. 10 illustrates a laboratory simulator which incorporates a complete vehicle braking system and W.S.P. hardware. The vehicle loading, dynamic weight transfer, wheel inertia and road surface adhesion are all simulated electrically. Visual and print-out information on the dynamic process is continuously available during simulated stops on uniform or varying adhesion surfaces.

ACKNOWLEDGEMENTS

We have been fortunate to be associated with the motor industry during this particular period of technological development and gratefully acknowledge the training, experience and assistance received from our colleagues.

We should also like to thank the directors of Girling, Limited and of Joseph Lucas Limited for permission to publish this paper.

Legal and Moral Considerations

Many of the worlds current braking regulations preclude the use of four-wheel anti-locking systems. This is hardly surprising, since most regulations were put together long before the development of such systems was envisaged. It would therefore be helpful to remove these legal obstacles and thereby encourage vehicle manufacturers to adopt four-wheel anti-locking systems.

An interim stage, which would also be most helpful in accumulating service experience of such complex electronically-controlled braking systems, is the introduction of rear-wheel anti-locking systems.

Such systems would not create legal problems and their performance, even in the various failed modes, would never be worse than a standard braking system. This would also provide an incentive to develop the necessary servicing facilities, including diagnostic equipment for plug-in checking of the electronic control circuits.

One of the main moral questions surrounding anti-locking systems concerns the electronic 'black box'. Is it reasonable for an electronic circuit to make independent decisions on behalf of all drivers on the basis of a preconceived concept of what is best in the majority of situations? As a point of law, it would also seem that a system which can–in certain circumstances–be inferior to standard brakes might be construed as indefensible?

The fail-safe characteristics and ultimate reliability of such anti-locking systems will have to be of the highest order if they are not to cause numerous additional accidents.

CONCLUSIONS

The conventional hydraulic braking system has provided an acceptable level of performance and reliability over many years, but improvements are possible to create safer braking systems. Every improvement involves additional braking components and the ensuing complexity may lower the system reliability, thus inviting additional brake failures, although with less disastrous risks.

Commensurate with the increased failure risks will be a need for better maintenance. Sophisticated diagnostic servicing facilities may become essential to maintain optimum performance in the field.

The recent introduction of numerous international braking regulations has focused the attention on safety and accelerated the development of new products and systems; such as divided circuits, warning indicators, apportioning valves and anti-locking brakes.

Accident studies are necessary to evaluate the cost-effectiveness of such devices.

The envisaged application of electronic circuits to control the braking system under emergency conditions, without reference to the driver, poses both legal and moral problems.

APPENDIX

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ACCIDENT AVOIDANCE SEMINAR

SECTION THREE PART 2

STEERING, HANDLING AND BRAKING

The following report (printed verbatim) was submitted for inclusion in the seminar minutes by Daimler-Benz AG.

THE INFLUENCE OF ROAD CHARACTERISTICS (SLOPE, INCLINATION, BENDS), OF THE ENGINE-BRAKING MOMENT AND OF THE ROTATING MASS ONTO THE BRAKE-FORCE DISTRIBUTION AND ON THE BRAKE-CIRCUIT SPLITTING

DR. MANFRED BURCKHARDT
Daimler-Benz AG
Federal Republic of Germany

The problems of road-vehicle brakes are focussing the general interest from time to time. Again and again new concepts are brought forward and discussed. Comprehensive theoretical work is being achieved and published–frequently with Government assistance. Popular periodicals enthusiastically welcome this issue and give rise to ardent discussions at the end of lectures.

This work has been closely watched by us and in doing so it has become evident that insufficient attention of quite a number of substantial angles has been overlooked. It thus appears to be appropriate to also bring forward some arguments on the subject of rating vehicular brake systems for discussion.

At the outset there is the question: What is properly understood by modern brakes? We feel that this question should be answered as follows: Modern brakes are of simple construction and of extreme dependability. They refrain themselves from any useless complexity and they are so rated that they stop the vehicle even under the most adverse conditions (as, for instance, with transverse acceleration or on a downgrade) without more detriment to handling stability than avoidable physically.

This definition agreed upon, the result of a meticulous, however not simple analysis is that the two-circuit splitting into front and rear axle circuits – regarded as no longer corresponding to the state-of-the-art by some popular articles – with fixed distribution – that is without the so-called brake-force distributor – yields in a quite modern brake as long as its distribution is performed carefully and factors of influence the effects of which are to be shown in this presentation have been taken into due regard. Rating of the entire design is based on an occupation with the matter for decades and it is finally condensed in subtle considerations and accompanied by some thousand brake tests.

As a member of a Company the vehicles of which are throughout equipped with a brake system of the described construction, I am highly pleased to lift a little the veil hanging over our research and development work. I feel particularly indebted to the Board of Messrs. Daimler-Benz AG who kindly gave the permission to do so.

This short lecture will, as I hope, reveal the considerations which among others have contributed to our rating of the brakes and at the same time they will make evident that there is no reason to deviate from our principles which have stood their proof, even when a superficial observer might possibly come to the contrary impression. There is, however, no doubt that we will not hesitate introducing a superior system should we be able to find one.

Our brake investigations concern so-called conventional vehicles: front engine and rear-axle drive. Other types of vehicle construction as, for instance, front-axle drive, result in entirely different achievements. As revealed by these investigations, the conventional principle of construction possesses the best all around characteristics for engines of more than 120 to 130 hp. For lower engine outputs, a vehicle with front drive might have merit.

A large percentage of the vehicles of our company disposes of outputs in excess of 120 hp so that the general use of the conventional vehicle construction is, in toto, to advantage and for this reason I may be permitted to limit myself to the analysis of the thus resulting brake performance.

Rating of the brake system is based on the parabola of the ideal brake-force distribution the derivation of which is certainly known to this audience. The locus of the center of gravity of the vehicle as to the coordinates of its height and length including the wheelbase has been taken into due consideration.

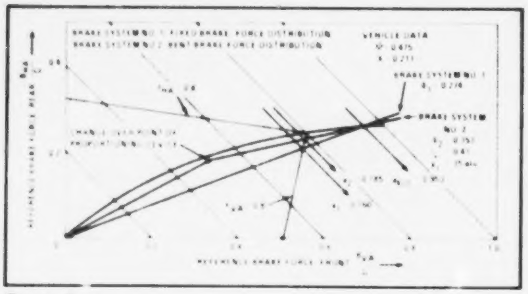

Figure 1

Among others, Figure 1 shows such a parabola of the ideal brake-force distribution, supplemented by the lines of constant retardation and the lines of constant coefficient of friction of tire to road surface. In the area above the parabola, first the rear axle locks whereas in the area below, it is the front axle locking first whenever the friction at the pertinent axle is fully utilized. The ideal brake-force distribution cannot be realized at all.

For this reason, a technically feasible rating is incorporated into the vehicle which we call the "installed brake-force distribution." Lockup will not occur until the curve of the installed brake-force distribution intersects the actually pertinent line of constant coefficient of friction of the tires. The line of constant retardation intersecting this point is a measure for the vehicle retardation so obtained. When the driver stronger depresses the brake pedal while the front axle first locks, the actual brake-force distribution moves along the line of constant coefficient of friction until the parabola of the ideal brake-force distribution has been reached. In this point, the second axle, too, is locking which simultaneously induces the possible maximum retardation. This remark is only valid under the assumption that the coefficient of friction does not change its magnitude even during lockup but keeps its once determined magnitude. In the field this is not the case. In order to come to transparent relations, we are going to continue the assumption of a constant coefficient of friction.

It may be recalled that the line f_{HA} = constant passes through the point / on the abscissa (= share of rear-axle load, = relative height of center of gravity), whereas the line f_{VA} = constant passes through the point –(1 –) on the ordinate. For the sake of a clearer presentation, the lines $f_{HA} = 0.8$ and $f_{VA} = 0.8$ have been entered only into the diagrams.

Two installed brake-force distributions form the base of the subsequent considerations, that is–

(1) a fixed distribution with a point of intersection with the ideal brake-force distribution at $a_{crit.} = 0.953$ and

(2) a broken distribution with a pressure-dependent distributor (sometimes, though not correctly, called brake-force regulator) with a changeover pressure of about 35 atü and a step-down ratio of $i_v = 0.41$. In this instance $a_{crit.}$ = is likewise 0.953.

Apparently both ratings are stable since in both instances for a coefficient of friction of 0.8 the front axle is first to lock yielding in retardations of a = 0.76 for the fixed distribution and of a = 0.785 for a pressure-dependent distributor.

As soon as the brake pedal is stronger depressed, the actual distribution travels along the line $f_{VA} = 0.8$ up to the intersection with the parabola of the ideal brake-force distribution. The resulting maximum deceleration then obtainable is identical in both instances.

The shown diagram with its clear statements is valid for a stop on a level road and with disengaged engine. In case of high powered vehicles which increasingly are equipped with automatic transmissions, however, the diagram calls for supplementing. While it is reasonable to assume that a driver of a vehicle fitted with a manual gear shift will disengage the clutch during a critical brake application, an action on behalf of the driver cannot be taken into account for vehicles with automatic transmissions. Thus, the influence of the rotating mass of the engine and of the engine-braking moment onto the brake-force distribution calls for an investigation. (This will also hold true for vehicles with mechanical transmissions not disengaged by their drivers.)

The result is shown in Figure 2. Let us first consider the fixed distribution. The brake of the engine at the rear wheels, transformed into the diagram of the brake-force distribution, yields in a curve the shape and position of which is a function of the driving speed and of the engaged gear. The influence of the engine is the greater, the greater the speed of the vehicle, the lower the gear and the smaller the coefficient of friction is. The deviation from the installed brake-force distribution is small with high retardations, however, increases materially towards smaller decelerations. This counts for the well-known fact that swaying of the rear axle can be induced solely by an abrupt zero positioning of the accelerator in the instance of high powered vehicles and slick lanes (as, for instance, with ice or snow).

The thus resulting displacement of the installed brake-force distribution leads to a reduction of

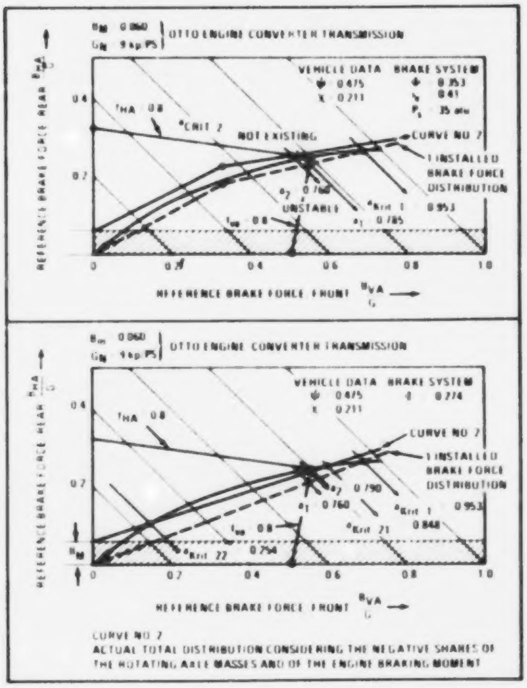

Figure 2

$a_{critic.}$ to 0.848 in case of a fixed distribution. The front axle locks for f = 0.8 at a = 0.79. The stopping distance is therefore slightly smaller than without the engine. Below $a_{critic.}$ = 0.254 corresponding to coefficient of friction of tire/road surfaces of identical magnitude the vehicle becomes unstable. Between a = 0.848 and a = 0.254, however, the ideal brake-force distribution is rather well approximated.

For a vehicle featuring a pressure-dependent distributor, it at once shows up that this one has become unstable throughout the entire operating range. With a coefficient of friction of f = 0.8, first the rear axle locks obtaining a deceleration of a = 0.76.

Summing up, one can state that for a superficial consideration the fixed distribution seems to be definitely inferior in comparison to the brake-force in the instance of a straight-forward retardation, particularly in the middle range of friction with disengaged engine but actually when closely examined in the field which means with the engine, proves to be the clearly superior solution.

Unfortunately, a vehicle is not only decelerated on level ground but also on downgrades and uphills. The latter case is of no interest as soon as the starting point is a proper distribution for level-plane stops since the braking stability is anyhow improved in comparison to the level ground. What, however, is the situation on downgrades?

In order to make this situation more transparent, comprehensive theoretical investigations and driving tests have been performed which have referred to braking conditions deviating from the standard case, that is from the straightforward stop on a level ground.

The knowledge thus gained is of such an interesting nature that it has lead to a dissertation which just has been completed. (Dissertation of Egon-Christian von Glasner entitled "Distribution of Vehicular Brake Systems", Technical University of Stuttgart, Germany). A considerable portion of this paper is based on this fine work.

Figure 3 shows the effect of upgrade and downgrade onto the ideal brake-force distribution. First of all, we have to differentiate between the reference brake force a* and the retardation a which are congruent when considering the level-ground situation and when neglecting the air forces. By "a*" is indicated which fraction of the weight is effective as a braking force at the axle or at the vehicle while "a" states the actual retardation, that is the retardation with reference to the gravitational acceleration.

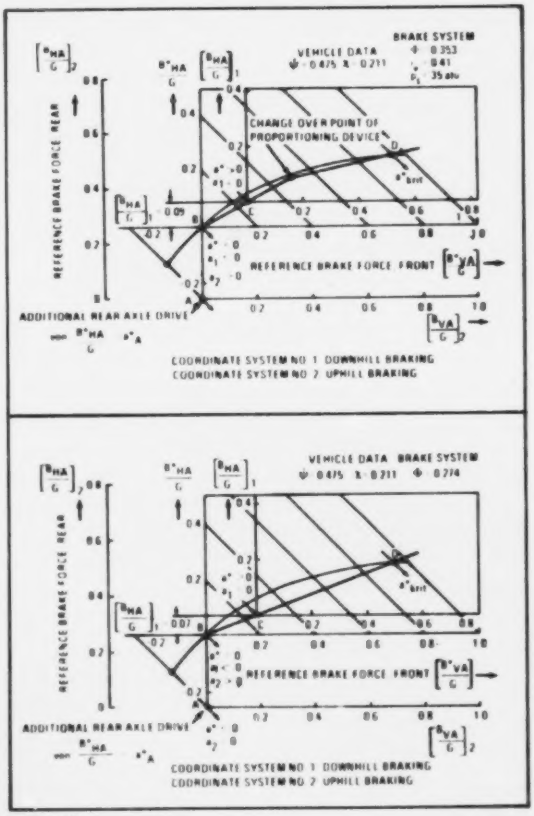

Figure 3

As long as the (tangential) forces transferred by the wheels only act on the vehicle, a* and a are equal. Air forces or drive through the downgrade change a in comparison to a*.

The influence of the downgrade is suitably represented by a displacement of the zero point of the system of coordinates for a retardation along the installed brake-force distribution. Due to the steeper rise of the installed brake-force distribution with a pressure-dependent proportioning device, the upward displacement (of 0.09 in the example) is greater than for a fixed distribution (of about 0.07).

For representation of the influence of the upgrade, the zero point of the system of coordinates, as can easily be shown, is displaced downwards assuming rear-axle drive as stated at the outset. Retardation a as well the stability of the vehicle are being considerably improved for a given a*.

In Figure 4, a retardation downhill on a slope of 15 percent is shown together with the influence of the engine retardation (for a given speed). It becomes evident that for a vehicle with a pressure-dependent distributor and a coefficient of friction of f = 0.8, solely a retardation of a = 0.58 can be obtained in which instance the rear axle will lock first. The vehicle with fixed rating is stable in this instance and permits a maximum retardation of a = 0.627.

Much could be said to this issue but the limited time available compels changeover to the next item.

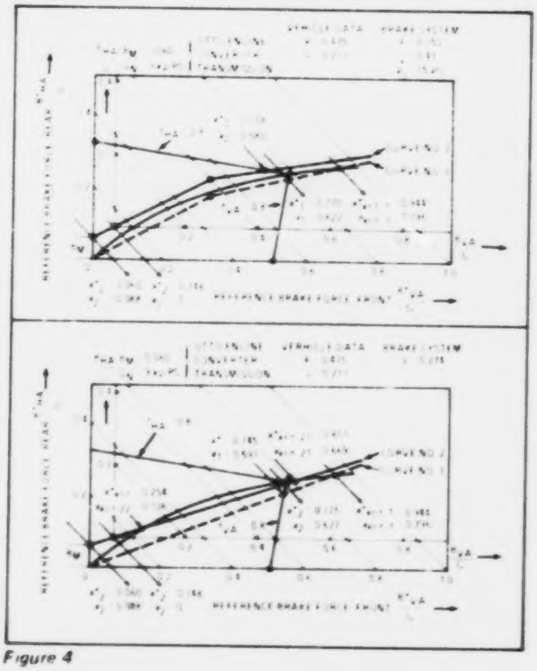

Figure 4

Besides a straight-forward retardation on downgrades and under due consideration of the influence of the engine which, as has been shown, is substantially deviating from the customarily considered sole straight-forward stop on a level ground, there is no doubt that retardation in a curve is of primordial importance under todays traffic conditions. It often happens, particularly on speedways, that a retardation is imposed onto the driver in spite of the fact that his vehicle is subject to a substantial transverse acceleration during the simultaneous ride through a bend. The following data may illustrate the situation:

With a radius of the bend of 100 m (328 ft), the transverse acceleration at a speed of 72 km per hr (44.7 mi. per hr) amounts to 4 m per sq sec (13.1 ft per sq sec). In the instance of a speed of 180 km per hr (112 mi. per hr), a radius of 1,000 m (3,281 ft)–that is a faint curvature only–suffices already for a transverse acceleration of 2.5 m (8.2 ft per sq sec) per sq sec.

By means of the brake-force distribution diagram–that is by equivalent means as with the downgrade retardation–one can obtain the lockup point of the most critical wheel for a given installed brake-force distribution and a definite coefficient of friction. In doing so, one brake-force distribution diagram has to be established for every transverse acceleration. This, of course, will not be done actually but the problem is programmed and treated by an electronic computer.

When establishing the equations, use can be made of the so-called two-wheel vehicle according to Riekert and Schunk as has also been done by Wallner at Professor Fiala's chair. However, the displacement of the wheel load of the four-wheel vehicle has to be introduced via the instantaneous center of the axle whenever unpermissible errors are to be avoided. The stability consideration is then performed for the "corrected two-wheel vehicle" of the road wheels adjacent to the curve which in our experience are decisive for the reaction of the vehicle. A more detailed discussion of this issue, however, is too time consuming and reference should be made therefore to the above mentioned dissertation.

When plotting the results of the described calculation for the vehicle on which this consideration is based, Figure 5 is obtained. Here, the available retardation of the vehicle up to lockup of the most critical road wheel–without consideration of the engine braking moment–is entered as a function of the transverse acceleration. Graph No. 1 is valid for the already known vehicle with fixed distribution while No. 2 represents the vehicle with a pressure-

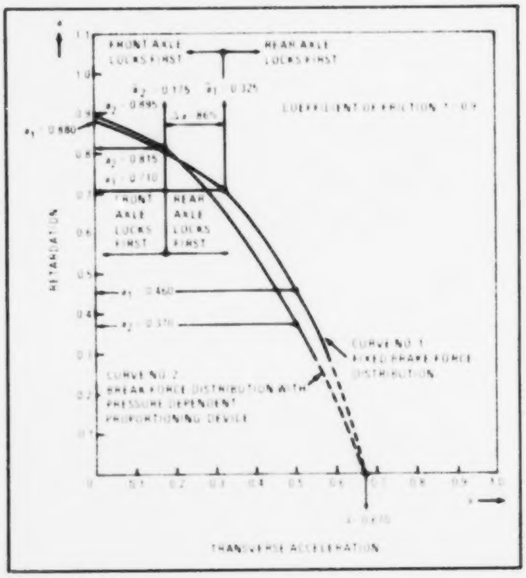

Figure 5

dependent proportioning device of the already described characteristics.

It can be taken from the diagram that the vehicle with fixed distribution can obtain quite large retardations even in the instance of relatively high transverse accelerations without rendering the vehicle non-maneuverable. On the other side, the vehicle fitted with a pressure-dependent proportioning device becomes unstable at considerably lower efforts. In the diagram, the so-called critical points are indicated at which a transition from overbraking of the front axle to overbraking of the rear axle is taking place. From this, it can be seen that the vehicle with the pressure-dependent proportioning device turns to instability already above a transverse acceleration of $\bar{a} = 0.175$ at the rear axle while this condition does not take place with the vehicle of fixed installation above $\bar{a} = 0.325$. What is more important for the average driver is the fact that the vehicle fitted with a pressure-dependent proportioning device experiences swaying his vehicle for a transverse acceleration of $\bar{a} = 0.5$, which results for a fast drive already with rather large bend radii and a retardation of $a = 0.37$ at the rear axle. The vehicle equipped with the described fixed rating still permits a retardation of $a = 0.46$. The vehicle with the fixed distribution, therefore, shows a considerably lower "brake-induced rotating effect."

Some time ago, a new term has been created which, in part, has been called "braking efficiency," in part, differently. This term has even been proposed for introduction into pertinent legislation.

When comparing our above result, we come to the conclusion that just the superior vehicle shows a poorer "braking efficiency." Vice versa, the vehicle with the better "braking efficiency" (that is the one fitted with a pressure-dependent proportioning device) has a markedly inferior brake performance in bends.

We thus come to the question if there is any meaning or even responsibility when determining the "braking efficiency" by rules and regulations. Such a measure could induce a manufacturer to designing of vehicles—against his better knowledge—the brake performance of which, for instance on downgrades, with engaged engine and in the bend, is materially worse than necessary.

This only done in order to correspond to a law which has come to light solely by theoretical considerations which refer to straight forward driving on a level ground and with disengaged engine.

Unfortunately the described laws on the brake-force distribution, theoretically to be installed for series production, are extremely difficult to describe. The actual brake-force distribution often deviates from the installed brake-force distribution in the field. This is due to the brake linings. All of the shown considerations are based on the assumption that the coefficient of friction between brake disc and brake pad corresponds to the mean value on which the calculation is based in all instances. Actually, however, this is not the case.

The expert knows that the coefficient of friction of a brake pad undergoes a considerable scatter even in case the exterior conditions are kept constant. By many thousands of tests we have shown that there is one Gauss error-distribution curve for every brake condition when a sufficient number of measurements has been performed. Figure 6 illustrates such a result. The frequency with which the mean coefficient of

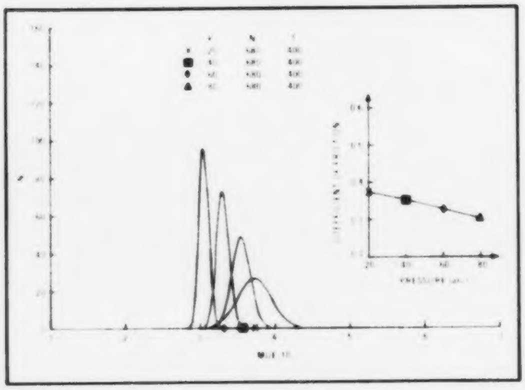

Figure 6

friction occurs for this couple of disc/pad when the exterior conditions (brake pressure, initial velocity, initial rotational speed, initial temperature) are kept constant, has been plotted vs. the mean friction coefficient as the abscissa.

It becomes evident that there is a different Gauss distribution for every brake pressure. Entering their maxima into a diagram (right-hand supplementary diagram) shows the dependence of the coefficient of friction from the brake pressure.

A similar dependence is found when varying the intial temperature (Figure 7) or the initial speed (Figure 8). Reviewing all of these results, it seems to be almost impossible to come to a useable layout for a vehicle in view of so wide a scatter. It should be further kept in mind that thought has been given here only to the mean coefficients of friction which can be calculated exactly, for instance, from the braking time of a defined flywheel mass. The instantaneous data are frequently of an even broader scatter.

The shown large differences of the coefficients of friction cannot be compensated by any brake-force proportioning device whatsoever.

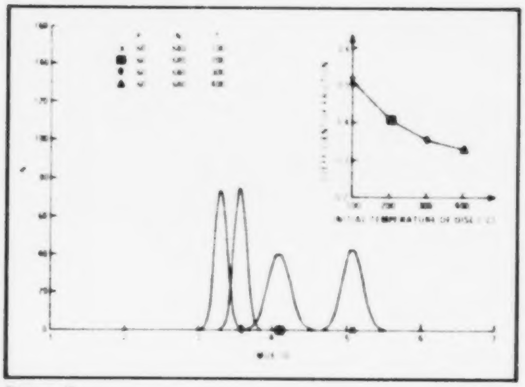

Figure 7

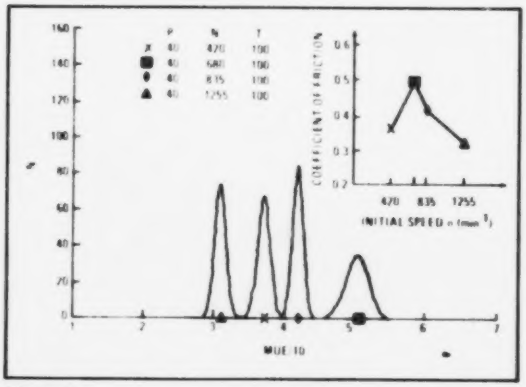

Figure 8

In order to ensure a stable vehicle performance under all practical conditions, other means have to be used which shall be briefly explained.

First of all, there have to be testing facilities which permit exact simulation of the conditions in the vehicle. Their results should be rapidly and unmistakably evaluated. In our case this is a flywheel-type test bed in which half of the vehicle mass is retarded by the front-axle and rear-axle brakes on a common shaft, fitted with original equipment and original controls (master brake cylinder, tubing, brake calipers). The moment occuring at the brakes is measured and handled by a small analog computer such that it can be directly entered into the previously calculated and plotted brake-force diagram of the vehicle under consideration by means of an XY recorder.

The result is shown in Figure 9 in which the course of the brake forces for constant induced pressure is plotted. The shown couple of disc and pad – front axle to rear axle – has been optimized by hundreds of tests. Exact matching of the course of the coefficient of friction of the pad in view of the developed temperature at the brake disc is required. By the larger amount of absorbed energy the disc temperature at the front axle increases more rapidly in spite of the somewhat greater disc mass than is the case at the rear axle. Above a definite temperature, the pad of the front wheel tends to reducing its coefficient of friction. For this reason, the pertinent rear wheel pad should for the corresponding temperature, which is materially lower, likewise display a reduction of the coefficient of friction of a magnitude compensating for the decay of the coefficient of friction of the front-wheel pad.

This matching is rather troublesome although, in our opinion, it represents the sole correct answer to

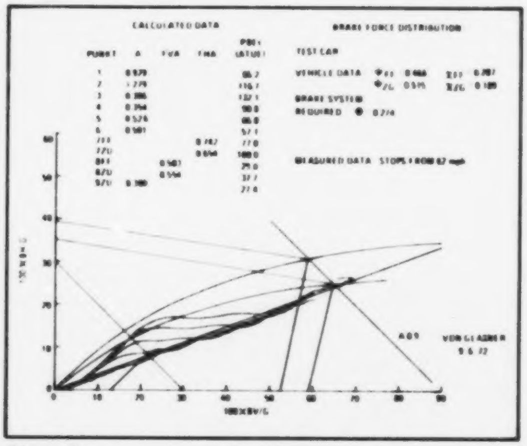

Figure 9

the problem. With a careful and homogeneous production, the course of the coefficient of friction once determined is reproducible. To this end, the manufacturer of the brake lining has to make substantial efforts.

The large absolute scatter of the coefficient of friction is preserved. However, it can so-to-speak be neutralized by a proper matching.

The momentary overemphasis of the rear-axle brake force during the initial brake phase, that is at the outset of the braking cycle, can be traced back to the friction of the floating plunger in the master brake cylinder.

What happens when the indicated conditions are not adhered to is shown in Figure 10. In this instance, commercial pads have been installed the average coefficients of friction of which resulted in the plotted installed brake-force distribution obtained by calculation.

The brakes then are warmed up in accordance with a schedule which, for instance, corresponds to a rapid downhill drive at the "Zuflucht," a mountain course of the Black Forest near Freudenstadt. It becomes evident that the brake-force share of the rear axle is much too large. Equipped with these brake pads, the vehicle would directly present a safety hazard since it is unstable. This is a particularly fine example for demonstrating the degree of change of the braking performances of a vehicle by the use of brake linings the initial and/or nominal coefficients of friction, it is true, of which are correct the performance of these coefficients of friction, however, does not comply with the requirements. It should, therefore, be urgently demanded to install only such friction pads at front and rear axles as authorized by the manufacturer of the vehicle. If this conditio sine qua non is not adhered to strictly, a serious safety risk is inevitable.

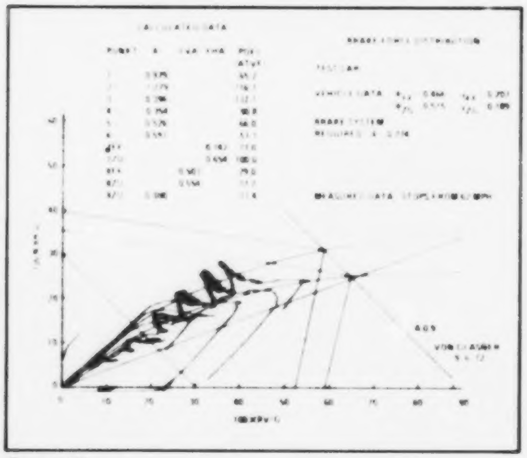

Figure 10

The distribution of the braking energy onto front and rear axles corresponds to the installed brake-force distribution. This results in a higher energy absorption of the front brakes than of the rear brakes. For design reasons there are limits to enlarging the mass of the front brakes to the same extent. This, in turn, will as a rule cause their substantially higher heating up than with the rear brakes.

When the brake system is in order, the brake fluid sufficiently temperature-resistent and the driver reaction all-right, the above statement is inconsequential.

However, should the brake fluid, and extremely hygroscopic polyglycole, have absorbed water and the boiling point thus might have been lowered considerably, critical conditions result. This is particularly so since the driver is not in a position to evaluate which driving conditions are risky ones. Besides, he mostly could not use such knowledge due to the traffic situation.

Terrible vaporization of the brake fluid at the most critical spots will take place. Under unfavorable conditions, build-up of pressure is no longer possible in the pertinent brake circuit. The brake then fails in this circuit.

With a two-circuit splitting into front-axle and rear-axle circuits, this can be tolerated provided driving habits are appropriate. The higher stressed circuit, that is the front axle, fails. The driver will notice a material lengthening of the pedal travel and a proportional reduction of the braking effect which may suffice when the distance to the preceding vehicle is reasonable and correct. He is warned and can stop the vehicle by the rear-axle circuit.

This is not so with all kinds of two-circuit brakes in which both circuits act upon the front axle (for instance, by four-cylinder calipers). Is vaporization experienced here, then the vapor of the brake fluid will be emptied on occasion of the next discharge of the brake pedal that is by uncovering the "snuff bore" in the brake cylinder so that no pressure can be built up during the next actuation of the brake pedal. Thus, both circuits of the service brake have failed simultaneously and the vehicle can be stopped only by the parking brake.

This undesirable effect is the primary argument for our adherence to a brake-circuit splitting into front and rear axles which often is considered as not representing the state-of-the-art.

There are, of course, quite a lot of other reasons which cannot be discussed at present because of time shortage. Reference to the publication on "Contributions to the problem of the two-circuit brake," ATZ 1969, Number 7, may suffice instead.

The above discussion seems to be of particular importance in view of the legislation. Neglection of the temperature development and of the vaporization might possibly lead to designs which theoretically seem to be safe but are objectively rated as poor. Unrealistic data for determination of the required retardation in case of circuit failure which, for instance, could be complied with only if both circuits were acting onto the front axle might perhaps compel vehicle manufacturers to design contrary to their better knowledge. And this only for the sole reason of meeting retardation values required by law. The safety measures against failure of brakes by vaporization would, as has been shown, be substantially reduced so that the end result would mean an unsafe brake system.

May I express my hopes that some problems of brakes have comprehensibly been presented in spite of the limitation to few details and the double-quick step necessitated by the short period of time available for the purpose. I hope as well that I might have succeeded in demonstrating the degree of attention paid to these problems by our Company. This meticulous care is acknowledged by our customers which are satisfied with the brakes of their vehicles. We would be highly pleased if some of the presented thoughts would invite attention of experts.

ACCIDENT AVOIDANCE SEMINAR

SECTION **THREE** PART **2**

STEERING, HANDLING AND BRAKING

The following report (printed verbatim) was submitted for inclusion in the seminar minutes by Dainler-Benz AG

HOW GOOD IS THE BRAKING EFFICIENCY AS A CRITERION FOR EVALUATION OF A BRAKE SYSTEM?

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1. Nomenclature, Dimensions And Indices

a	retardation $= \frac{\text{deceleration}}{\text{gravitational acceleration}}$	
f	coefficient of friction, tire and road surface	
g	gravitational acceleration	m/s^2
s_i	ideal stopping distance	m
s_r	real stopping distance	m
t_u	pressure build-up time, loss time	s
$\frac{B_{VA}}{G}, \frac{B_{HA}}{G}$	braking forces, front and rear, referred to vehicle weight	
$F(x)$	function of (x)	
ϵ	utilization of coefficient of friction. ("Braking Efficiency")	
ψ	rear-axle fraction $= \frac{\text{rear-axle load}}{\text{vehicle weight}}$	
χ	relative height of center of gravity $= \frac{\text{weight of center of gravity}}{\text{wheelbase}}$	
Φ	rear-axle brake-force fraction $= \frac{\text{brake force at rear axle}}{\text{total brake force}}$	
μ	coefficient of friction, brake lining	

Indices And Abbreviations

VA	front axle	FF	ready for operation with full tank and driver
HA	rear axle	ZG	permissible total weight

2. Introduction And Definition Of "Braking Efficiency"

A term often encountered recently, is the so-called utilization of the coefficient of friction, also erroneously called "Braking Efficiency." The utilization of the coefficient of friction is to demonstrate which percentage of the available coefficient of friction may be utilized up to lockup of one axle for the retardation.

[4], [5], [6], [7], [8], [9], [10], [11], [12], [13], [14], [15]

Generally speaking, the degree of utilization of the coefficient of friction may be described by

$$\epsilon = \frac{a}{f}$$

A new definition of the "Braking Efficiency" has been given by R. W. Murphy and F. A. DiLorenzo [1], [2], [3]. To this end, the retardation and the coefficient of friction are replaced by the ideal and real stopping distances.

$$\epsilon = \frac{S_i}{S_r}$$

$$\epsilon = F(f, a, g, \nu_0, \psi, \chi, \Phi, t_u)$$

Since the ideal stopping distance depends on the coefficient of friction, the dependence of the coefficient of friction from the vehicle speed, particularly for small coefficients of frictions, is taken into account by this equation

$$f = A\nu_0^2 + B\nu_0 + C$$

For dry tracks, the ideal stopping distance has been specified as

$$S_i = \frac{\nu_0^2}{2 \cdot g \cdot f}$$

and for wet tracks as

$$S_i = \frac{A}{g} \int_0^{\nu_0} \frac{\nu_0 \, d\nu_0}{A\nu_0^2 + B\nu_0 + C}$$

In order to take into due regard the tire performance on dry and wet tracks, a standard tire is introduced and compared to the test tire.

The introduction of these comparative values and the determination of:

$$f = F(\nu_0)$$

for changing wheel loads entails a large measuring effort. In addition, track conditions have to be identically reproducible. Such reproducibility, in our opinion, is not at all times guaranteed, at least not for wet tracks.

It also cannot be taken clearly from variety of papers entitled "A procedure of evaluating vehicle braking performance" [1], [2], [3], whether or not this method is to be introduced for antilock systems only or for braking systems in general. While papers [1] and [2] deal with antilock systems, at least paper [3] leaves it open as to whether or not the shown procedure is generally valid or for antilock systems alone.

In both instances, factor ε underlying the evaluation of the vehicle, has to be examined closer. The arguments which do not support use of the "Braking Efficiency" ε are primarily valid for a regular brake system. When an antilock system has been installed, these arguments will also hold true since in case of failure of the device, the driver is supposed to have at his disposition an optimum brake system representing the present state-of-the-art.

The following criteria are crucial for the quality of a brake system:

1. as short stopping distances as possible under all track and load conditions
2. during brake application, the vehicle is to remain stable, that is under all track and loading conditions on a straight course and, possibly, also in curves.

These two criteria shall be analyzed in view of the "Braking Efficiency."

3. Discussion Of The "Braking Efficiency"

Short stopping distances mean high retardations. The retardation obtainable on a straight course is a function of many factors, essentially of vehicle parameters (rear-axle load fraction ψ, relative height of the center of gravity χ, rear-axle brake force fraction Φ) and of the coefficient of friction f between tire and track.

3.1. Significance Of The Term "Braking Efficiency"

3.1.1. Increase Of The Retardation In Excess Of Wheel Lockup At The Front Axle

As already mentioned, for calculation of the "Braking Efficiency" that retardation is employed which causes lockup at one axle. This retardation, however, is not identical with the maximum possible retardation since, in the instance of an earlier lockup at the front axle, the pedal force can be increased beyond the lockup point without any danger, thus obtaining a retardation corresponding to the coefficient of friction and by disregarding the coefficient-of-friction vs. slip performance of the tire. Figure 1 shows the ideal and an installed brake-force distribution as it is feasible for many passenger cars (line 1).

For a coefficient of friction of f = 0.6, the front axle locks up at a retardation of a = 0.56 (point a). By further increasing the pedal force along line 2 (f = 0.6), the parabola of the ideal brake-force distribution is met at point b. Here, the rear axle is additionally locked and at f = 0.6, the maximum retardation of a = 0.6 is obtained. In this point ϵ equals 1. The possibility of further increasing the retardation in spite of front-axle lockup, is not taken into consideration by the "Braking Efficiency."

The maximum retardation at point b, however, cannot be obtained unless the vehicle remains stable during brake application, i.e., the rear axle is not to lockup earlier than the front axle.

The "Braking Efficiency," however, does not reveal the stability performance.

3.1.2. Stability Of A Vehicle

In Figure 1, for the vehicle loading condition "ready for operation with driver," an "unstable" fixed brake-force distribution has been plotted in addition (line 3).

For this, the "unstable" brake-force distribution has been so selected that it attains an identical retardation a = 0.56 up to lockup of one axle for a coefficient of friction f = 0.6 as the "stable" brake-force distribution, i.e. the "Braking Efficiency," is the same in both instances (ϵ = 0.93, see points a and c). While a = 0.6 can be obtained by the "stable" distribution, the vehicle with the "unstable" distribution begins to skid from a = 0.56 on as the rear axle locks up first.

Taking into consideration the course of the "Braking Efficiency" in the range from f = 0.2 to 0.8 of the coefficients of friction for both distributions, the "unstable" one, that is the unserviceable brake-force distribution, comes off extremely well. It even once attains the value 1 (Figure 2). This example does not only demonstrate the lacking significance of the "Braking Efficiency" regarding stability and retardation, but also the inappropriate denomination of a calculation value which is composed of so unsafe data as are μ, the brake-lining coefficient. Although disc brakes are much easier to evaluate in this respect than drum brakes, fluctuations of the coefficient of friction of ± 30% are also encountered with series brake pads of disc brakes. Footing on such an uncertain foundation, it is unrealistic to incorporate into a regulation the "Braking Efficiency" as evidence of a "good brake system."

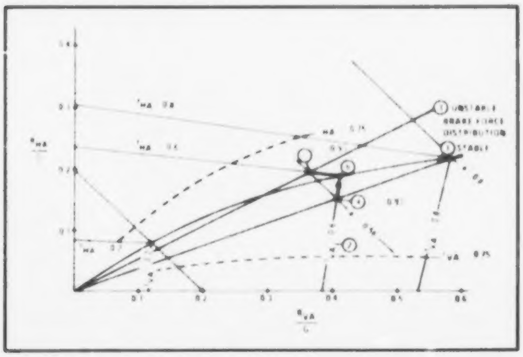

Figure 1

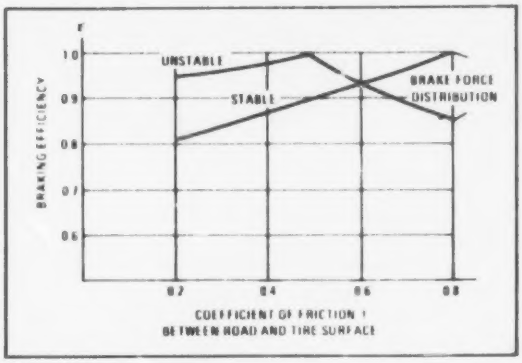

Figure 2

3.2. Major Factors Influencing The Design Of A Brake System Disregarded By The "Braking Efficiency"

When introducing a "Braking Efficiency" rating, there is no means of preventing its abuse for advertising purposes. This, in turn, would lead to obtaining an as good as possible "Braking Efficiency" when designing the brake system. By appropriate brake-force proportioning devices, it is even possible to achieve a "Braking Efficiency" of $\epsilon \geqslant 0.9$ throughout the specified range of coefficients of friction in which instance, in addition, theoretically the front axle locks up before the rear axle. An expert has to reject this brake system since he is to take into due consideration the effects described in greater details in subsequent chapters.

When planning motor-vehicle brake systems, fluctuations of the coefficient of friction, engine retardation moments and inertia moment of the drive, lateral accelerations, wind effects, drag of the vehicle and the like are to be taken into consideration and thus reserves for preventing overbraking of the rear axle have to be included in the calculations.

Some of these effects will be analyzed subsequently in brief and a survey in view of the "Braking Efficiency" be given on the comprehensive practical and theoretical investigations and findings of Daimler-Benz Company.

3.2.1. Fluctuations Of The Coefficient Of Friction Of Brake And Lining

The coefficients at the front and rear brakes may build up rather differently due to the different lining performances. Test results have shown that the brake force at the front axle may drop by more than 10% whereas the one at the rear axle may increase by the same amount (Figure 3 – transition of line 1 into line 2.) The rear axle is thus prone to lockup already from a = 0.61 (point a) on.

3.2.2. Effect Of Engine Braking Moment And Inertia Moments

The magnitude of the drag moment is mainly determined by the vehicle speed, gear ratio and engine size. In Figure 3, the effect of a six-cylinder engine at the third gear and for a speed v_0 = 50 km/h shown by curve 3. When the effects of the fluctuations of the coefficient of friction and engine-drag moment are superimposed (curve 4), it is distinctly shown that the rear axle is prone to overbraking in a large range. The effect of engine dragging cannot exactly be taken up during the planning phase. The proportional deceleration may vary from a = 0 (clutch disengaged) to a = 0.15 (first gear, upper end.)

3.2.3. Effect Of Vehicle Loading

In many passenger cars, the rear axle load increases with increasing load. This means that the ideal brake-force distribution shifts up, as shown in Figure 4.

Curve 1 is valid for "ready for operation, with driver" and curve 2 for the "permissible total weight." The fixed brake-force proportion (line 3) is not altered such that the reserve against over-braking of the rear axle increases. When this reserve is not used, the front axle locks up early as, for instance, for a coefficient of friction of f = 0.6 at a = 0.45 (point a). When further augmenting the actuation force, the ideal brake-force proportioning along line f = 0.6 is obtained and thus a maximum retardation (point b). The vehicle remains stable.

4. Summary And Conclusion

When practically designing a brake system, the passenger-car designer has to so select the brake-force distribution that a reasonable reserve against overbraking of the rear axle is maintained for all above described factors.

Vehicles of high driving performance have to be particularly equipped with a carefully matched brake system which shows up as a poor "Braking Efficiency."

This is the place where the argument could be heard that such vehicles had to be equipped with brake-force proportioning devices. Such devices, however, are only meaningful when their incorporation does not mean a loss of reserves against overbraking of the rear axle. By introduction of the "Braking Efficiency," all proportioning devices with a controlled changeover would be beyond discussion for conventional passenger cars for designers with a sense of responsibility.

This is so since these proportioning devices cannot dispose of sufficient reserves, in particular, in the critical range of small coefficients of friction and low rear axle load shares. Only load-dependent proportioning devices without a changeover point are serviceable in this respect.

The "Braking Efficiency" as a basis for evaluating of passenger cars is not only inappropriate but even dangerous since possibly a development can thus be

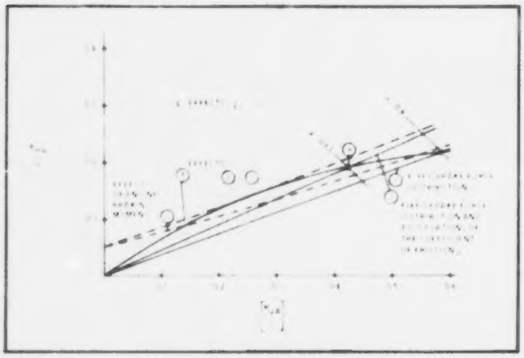

Figure 3

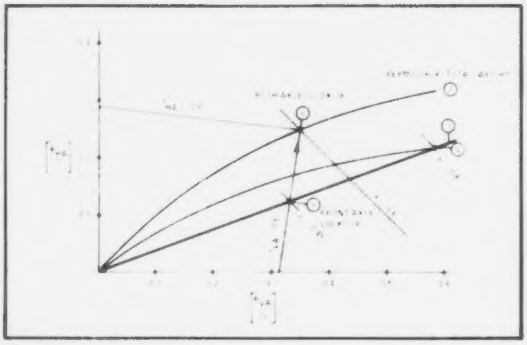

Figure 4

initiated which may lead to brakewise unstable vehicles.

The major argumentation for opinion shall once again be summarized:

1. The "Braking Efficiency" does not state anything on the actually obtainable retardations.
2. The "Braking Efficiency" is no criterion for the stability of the vehicle during brake application.
3. The calculation of the "Braking Efficiency" rests on unsafe assumptions for the calculation and is thus not suited for regulatory purposes.
4. The "Braking Efficiency" may be abused for comparative evaluation of vehicles which may lead to a dangerous competition.
5. A designer with a sense for responsibility faces a conflict in view of the "Braking Efficiency" because he can but show a poor one for reasons of stability.
6. Combined with the "Braking Efficiency," the introduction of brake-force proportioning devices do not result in a satisfactory brake system whenever the stability criterion is paid attention to satisfactorily.

A satisfactory evaluation of the obtainable retardations and of the stability performance of a vehicle for changing coefficients of friction, is permitted by the method proposed by M. Gauvin, representative of the French Government, for amending the ECE Regulation, in which with respect to the following equations

$$F_{VA} = \frac{B_{VA}}{G(\Lambda - \psi + a_x)}$$

$$F_{HA} = \frac{B_{VA}}{G(\psi - a_x)}$$

it becomes evident at a glance which axle first locks up for coefficient of friction (Figure 5). It is specified here that, for the range of f from 0.2 to 0.8, the curve of the front axle always has to be above the one for the rear axle vehicle which means that in this range the front axle always will lock up before the rear axle. Daimler-Benz Company supports this proposal.

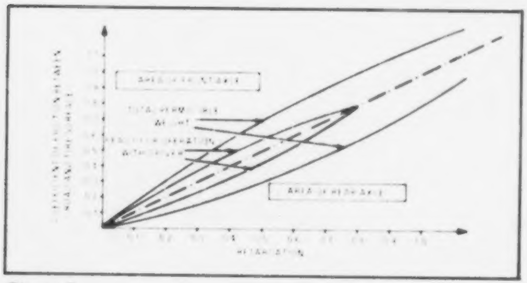

Figure 5

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ACCIDENT AVOIDANCE SEMINAR

SECTION THREE

PART 3

VISIBILITY, LIGHTING AND DRIVER ENVIRONMENT

KIYOKAZU YOKOI
Japan Automobile Research Institute
Japan

Ladies and gentlemen, I am with the Japan Automobile Research Institute and I am in charge of the visibility test of the ESV. At present, there are some problems regarding visibility and, therefore, I would like to present my statement on the reliability of visibility. The first is related to direct and indirect visibility under various weather conditions when we want good visibility with wipers and other devices. The visibility is closely related to crashworthiness. Sometimes you have to have a large sail panel or large pillars and there should be a good balance between the size of these devices and the visibility. The second is related to the lighting and signaling. In the ESV our objective is to increase the lighting intensity, keeping in mind that when the intensity of light is increased, there is the disadvantage of additional glare to the cars in front and to the rear and there is a problem in obtaining good balance between the glare and the intensity of light. As the trailing lamps have been developed by various companies, like Nissan, some problems still remain waiting for solutions. When you are to use new signals, there should be a uniformity on a worldwide basis. Therefore, in developing new trailing lamps and other lamps, we have to have utilization by all the countries in mind. This can be considered as one problem.

The last is related to the instrumentation panel. In this field, more research should be done on the motion of people, since we do not know much about human factors in this area. We have done a lot of research in the field of vehicles, but I think more human factors should be studied.

Regarding the drivers' environment, all the ESV manufacturers are now making efforts and there aren't common problems among all the manufacturers in the drivers' environmental problem solution. With this I'd like to complete my presentation.

Mr. Boschetti: I am George Boschetti of Peugeot, France. The first Japanese speaker indicated that it was necessary to make modifications in lighting problems to take into account the international rules and regulations, and he is quite right. I would like to say that, for future progress in particular, in ESV studies we cannot always remain under the existing rules and regulations. The Vienna Convention is the one that rules at this time, but there will certainly be a new meeting so that this problem can be brought up-to-date. For example, a certain number of manufacturers have proposed different rear lights. In our company we have proposed rear lights which we hope will prevent accidents when there is heavy traffic by cars driving at very high speeds. We believe that the level of speed on the roads is a very important indication of safety measures to be taken, particularly when traffic jams are formed, whether this be because of an accident or just a traffic jam.

Different colors will be visible from a very great distance, and I think that this will certainly be great progress as far as safety is concerned. This question of long distance visibility is important because the car is not an engine which should idle along, but, to prevent an increase in accidents, cars should be driven at what we call a correct speed. This speed depends entirely on the structure of the car, the quality of the driver, and also on the indications that are given him. One of these indications is the overall speed of all the other cars on the auto routes.

Mr. Clavel: My name is Maurice Clavel of Citroen. The Japanese report mentioned the problem related to the instrument panel and the driver, and we are wondering if this is really a question for ESV or just a question for standard cars. We must not confuse these two and we shouldn't stop progress by standardization, because any standardization will, of course, keep the manufacturer at a certain status–and he needs to go ahead and make progress. We ought to be able to help the drivers in this sense by producing cars which are simpler, but at the same time certainly much safer. So, we believe that to stop progress by standardization would be completely against safety. Our ideal is as far as the driver is concerned, that he should be able to direct his car in any manner whatsoever and that the car should be stable under any conditions.

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The second point concerning the instrument panel and the visibility of these panels is that everything should be done so that the control instruments and the panel which he is watching most frequently will be visible by the driver in an indirect manner so that he does not have to take his eyes off the road.

SECTION THREE

PART 3

VISIBILITY, LIGHTING AND DRIVER ENVIRONMENT

MR. HAMMERSTEIN
Adam Opel, AG
Federal Republic of Germany

I've been asked to give some comments, some explanations, on a new polarized headlamp system. This system is now developed in a manner that the efficiency of light output between the luminous source and the eye as a receiver is twice as high as it has been. Present polarizing sheets have a relatively low transmission factor of approximately 40%. You also need before your eyes an analyzer or visor and once more you have a transmission factor of only 40%. Together, they give you a 16% efficiency of your light output. We have found that with what we call a light divider, you can use two halves of the luminous beam; it is split up into parts, a reflected one and a transmitted one, and you now polarize both halves of the beam. You can apply this system to a 4-headlight system as well as to a 2-headlight system, and, in our opinion, it is possible to introduce it step by step; it's not at all necessary to change overnight the existing equipment of the entire car population in a certain country. You may begin with one vehicle, two, and so on, and you may add more vehicles equipped with polarized headlights every day. Because the system works like this you are provided a dip beam as well as a road beam. In the transition period where you have both systems mixed, today's system and the possible future polarized system, you treat oncoming traffic in the usual manner and you dip your beam if you see an unequipped car. One can recognize this car because the oncoming vehicle will dip and you can see that it is not equipped. As cars equipped with the new system increase in quanity, you can remain in the upper beam more and more. When you have oncoming vehicles, even in chains, and you are using the upper beam, the visibility distance is considerably higher than it is with the present dipped beam.

SECTION THREE PART 3

CONCLUDING REMARKS

DR. MASAICHI KONDO, *Chairman and Director*
Japan Automobile Research Institute
Japan

I would like to make some closing remarks. Had we more time I'm sure we could have had a more fruitful and active exchange of opinions. I am really sorry for the limited time. What I strongly feel now is the importance of human factors and clarification of human characteristics. These are my strong feelings.

First of all, we have to clarify the human characteristics and then we have to adapt the technology to this clarified human characteristic. Without this effort we'll never succeed.

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SECTION FOUR

CRASHWORTHINESS SEMINAR

PART 1 INTRODUCTION BY CHAIRMAN

DR. G. M. MACKAY
University of Birmingham
United Kingdom

PART 2 TECHNICAL PRESENTATIONS

DR. GENE G. MANNELLA
United States Department of Transportation
United States

ULRICH W. SEIFFERT
Volkswagenwerke
Federal Republic of Germany

PHILIP VENTRE and J. PROVENSAL
Renault
France

TERUO MAEDA
Nissan
Japan

R. DUNCAN LISTER
Transport and Road Research Laboratory
United Kingdom

VITTORIO MONTANARI
Fiat
Italy

YOSHIO SERIZAWA
Nissan
Japan

COLVER R. BRIGGS
Ford Motor Company
United States

DR. MAX DANNER
Accident and Motor Traffic Insurers (HUK)
Federal Republic of Germany

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RAYMOND G. FISCHER
General Motors Corporation
United States

EDWARD M. CHANDLER
United States Department of Transportation
United States

CRASHWORTHINESS SEMINAR

SECTION FOUR

PART 1

INTRODUCTION

DR. G. M. MACKAY, *Senior Research Fellow*
Department of Transportation and Environmental Planning
University of Birmingham
United Kingdom

INTRODUCTION

Good morning, ladies and gentlemen. I would now like to start the seminar on crashworthiness. My name is Mackay, from the United Kingdom, and I will be acting as the chairman for this discussion session. I would like to start by asking Mr. Manella, from the United States, to briefly begin the session.

SECTION FOUR PART 2

REMARKS ON 3,000 POUND ESV SPECIFICATION

DR. GENE G. MANNELLA, *Associate Administrator for Research and Development*
Research Institute
National Highway Traffic Safety Administration
United States Department of Transportation

I am not here in my capacity as Chairman of the Fourth International Technical Conference on ESV. Rather, I have asked to make a statement at the seminar on crashworthiness because it would appear that a principal item of discussion will be the proposed goals and specifications for a 3,000 pound weight class ESV which we forwarded to you approximately one month ago. Although some of the comments I will make were reflected in Mr. Scott's talk, I think they are important enough for a brief reiteration here.

We have had a number of inquiries regarding the possibility of open discussion and free exchange of comments on those specifications. You may recall that in our cover letter we suggested that this particular seminar would be a major forum for discussion of the proposed specifications.

I would like to briefly cover several major points that guided our effort in formulating these specifications. Recognizing that the ESV program is an experimental program, we chose to define goals rather than specifications, since the latter have the connotation of being rigorous numbers which permit no systems trade-off.

We set these goals high enough to stimulate creative engineering, yet at a level which was not patently unattainable. With regard to the technical approach involved, it is our intent to neither specify nor exclude any given technology.

I am certain that today you will hear several points of view on how to attain desirable crashworthiness at reasonable cost. Indeed, we would be delighted if participants in this seminar presented novel approaches to solving these problems.

A major point which we concluded from our experience with the first generation American ESVs was that acceptable crashworthiness levels without accompanying excessive weight penalties are the major technical challenge. This does not mean that we are unresponsive to other considerations of the vehicle-driver system, but that acceptable weights of ESVs would require a larger advance in the state-of-the-art than would the attainment of the other goals.

The increasing worldwide concern for energy conservation and growing urban congestion problems also suggest the need for smaller vehicles with adequate levels of safety. Accordingly, we view the 3,000 pound vehicle project as one where the crashworthiness weight trade-off will be the dominant portion of the effort.

Finally, we decided to circulate this document for comment and reaction in direct response to your request for involvement in setting the specifications for the ESVs. This is, in itself, somewhat of an experiment, and we will be listening carefully to your comments to see if constructive elements develop which can be factored into our plans. In other words, we have an excellent channel of communication regarding the next phase of this program. I hope that we can use it to our mutual advantage.

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CRASHWORTHINESS SEMINAR

SECTION **FOUR** PART **2**

SIMULATION OF ROAD TRAFFIC ACCIDENTS WITH BARRIER IMPACT TESTS

ULRICH W. SEIFFERT, *Department Manager*
Vehicle Testing-Research Center
Volkswagenwerk, AG

INTRODUCTION

In view of the fact that the head-on accident "front of vehicle against front of second vehicle" or "front of vehicle against fixed obstacle" counts as one of the most frequent types of accident encountered, attempts were made from a very early stage to simulate this type of accident by means of a direct frontal impact with a fixed barrier. In the early days of such testing, a wide variety of barrier types was tried out. Standardization of the barriers was due in no small part to the definition of barrier dimensions established as part of the work of the SAE (1). Since then, a number of national governments have declared the barrier test "impact at 90° into a flat plywood board attached in front of a concrete block" at 50 km/h to be a standard accident type, so that in the U. S. A., in Europe and in other countries, such as Japan and Australia, this procedure is used to assess the performance of a vehicle in the event of a collision.

The assessment criteria designed to provide information regarding the protection of the vehicle's occupants have, up to now, amounted primarily to limitation of rearward displacement of the steering and leakage of the fuel system during and after the impact test. In view of the fact that recently, in particular in the U. S. A. and Canada and in the course of work on ESV models, a change has been made involving the additional use of the frontal barrier test for assessing the restraint system, using test dummies, we can reasonably ask whether this test procedure is still compatible with the level of present-day knowledge.

Types Of Accident Occurring In Practice And Simulation Procedures

A number of investigations into the types of accident occurring on our roads have yielded information of a greater or lesser degree of complexity regarding the division of accidents on the roads into various categories. Basically speaking, accidents can be put into four categories: frontal, side, rear and rollover. The percentage breakdown of the accidents is approximately 50% frontal, 32% side, 10% rollover and 8% rear. In other words, some 50% of accidents comprising the types not falling into the frontal category cannot be simulated by the head-on barrier impact test. In view of the entirely different pattern of accident progress in the rollover accident, this type of accident has been disregarded in the remarks which follow.

For side collisions, two principal test methods are currently in use among those employed which simulate the impact with a rigid and non-deforming barrier at 90° (2), (3). This is defined in Standard 208 and in an ECE proposal. Within the program of ESV development, the vehicle is also run at 90° against a pole.

In the same way, a number of impact barriers have been laid down in specifications for rear-end collisions (ECE proposal, SAE, ISO), and these take the form of a completely rigid barrier which is propelled against the rear of the vehicle to be tested.

The problem of simulation with the aid of these test methods lies in the inability of the moving barrier to deform or in the fact that the test vehicle is moved into contact with a pole or fixed barrier. The moving barriers have, in addition to the drawback of non-deformability, the problem that they possess a given built-in mass.

As a matter of fact, only a very few road accidents are equivalent to crashing the vehicle against a non-deforming barrier. In an American survey (4), for example, we learn that 71% of all accidents involve a collision between two vehicles, and that a solitary vehicle occasions only some 21% of all accidents. A new study from Cornell in relation to fatal accidents shows that crashes against a pole are very often. These figures have to be proved. As we are aware, vehicles of widely varying mass are in motion on our roads at any time. Figure 1 shows distribution in the U. S. A. for the year 1971. We can clearly see that two peaks are present, namely at 1,800 pounds and at 3,600 - 3,800 pounds. In other words, the vehicles

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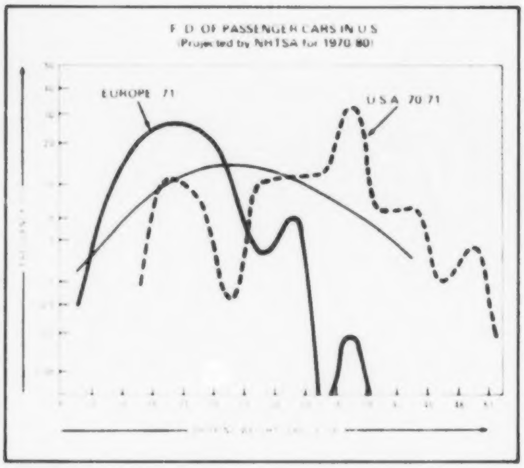

Figure 1

most frequently involved in a two-vehicle collision can possess a mass relationship of 1 to 2.

In addition to this varying mass ratio, we must recognize that the structures of the vehicles are different and thus variations in the force-deformation-distance pattern will occur.

If we attempt to simulate the head-on collisions actually occurring on the roads by means of an impact against a fixed barrier, the following shortcomings will become evident:

1. The relative masses of the two vehicles involved in the accident will not be taken into account.
2. The structure, that is, the deformation behavior of the various vehicles, will also be disregarded.
3. The inertial forces acting on the occupants are not the same for a vehicle-vehicle collision as for a vehicle-barrier collision.
4. Enhanced test requirements, for instance an increase in impact speed, can lead to unnecessary expense for the vehicle buyer without any effective benefit to him.

If we examine the individual types of accident we gain a somewhat clearer picture of the problems, which will now be described in detail with reference to the frontal collision.

The Frontal Collision

The frontal, or head-on collision, comprises all vehicle collisions taking place between 11:00 and 1:00 clockface angles against a fixed obstacle or against a vehicle also struck at this angle. Since such collisions are elastic-plastic processes, a common speed could be considered to have been attained after the deformation phase.

The following energy conversions take place:

$$m_1 \cdot v_1^2 + m_2 \cdot v_2^2 = \frac{1}{2}(m_1 + m_2)u^2 + \int_0^{s_1} F_1 ds_1 + \int_0^{s_2} F_2 ds_2$$

where:

s = movement of center of gravity

u = common speed after collision

v = speed before collision

The common speed u can be calculated

$$u = \frac{m_1 \cdot v_1 + m_2 \cdot v_2}{m_1 + m_2}$$

In the case of a frontal collision against a rigid barrier we can take mass m_2 to be infinite, speed of mass $m_2 \triangleq v_2 = 0$ and movement of mass $m_2 \triangleq s_2 = 0$. We then obtain the following expression:

$$\frac{m_1 \cdot v_1^2}{2} = \int_0^{s_1} F ds_1$$

In the barrier impact the total energy of the moving vehicle is transformed into deformation work. The barrier impact thus represents the simulation of a head-on collision between two vehicles of equal mass, speed and structure (symmetrical to the longitudinal axis of the vehicle).

The deceleration occurring in such a collision is a parameter which directly affects occupant protection. Disregarding the individual masses of the vehicle supported at the barrier, the deceleration provides a comparative measure for the differences exhibited by vehicles in respect of their deformation characteristic. Figure 2 shows a number of such deceleration patterns as a function of time, measured at approximately 30 mph on impact against a fixed barrier. We can clearly see that there are distinct variations in the deceleration/time level. We can assume, in greatly simplified form, that the measured deceleration multiplied by the vehicle mass yields the deformation force of the vehicle (this assumption can only be correct to a limited extent, since it does not take into account displacement of the mass center of gravity during impact). For vehicles with the engine in the rear this calculation can be made without a large error. For front-engine vehicles, where still greater differences in the force-distance level materialize, we are developing a computer program to calculate the forces which occur during a barrier impact. This can be seen in Figure 3.

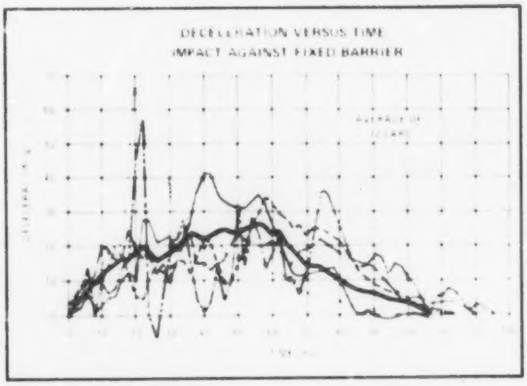

Figure 2

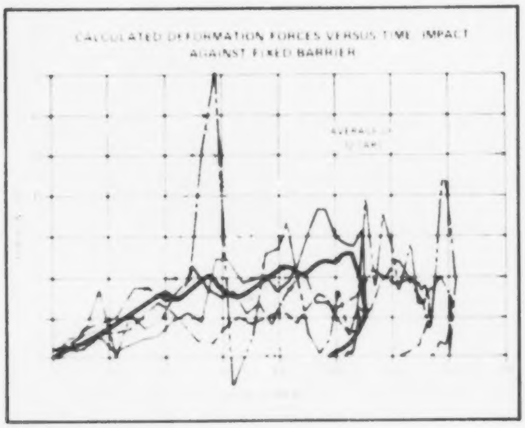

Figure 3

It was interesting to see from the investigation of the 32 different current production vehicles, which were classified in weight groups (1,300 1,760 pounds, 1,760 2,200 pounds, 2,200 2,650 pounds, 2,650 3,100 pounds), that the calculated average force-deflection characteristic was very simular.

Let us now demonstrate the head-on collision of two vehicles of different characteristics with the aid of a theoretical example. To simplify matters, linear force-distance patterns with, in some cases, a kinked characteristic curve will be assumed (Figure 4). A frontal collision at a Δ v of 80 mph would result in a deformation movement on the small (800 kg) vehicle of 1.055 m ≙ 40 inches, and on the large vehicle of 0.87 m ≙ 26.5 inches. The acceleration/time pattern for both small and large vehicles can be seen in Figure 5. This graph also depicts the impact of the small and the large vehicles against a fixed barrier at 40 mph. We can see that this deceleration-time pattern is not identical to that of the vehicle-vehicle collision, this being explained by the fact that in the case of the barrier impact each vehicle need only convert its own initial energy into work.

If one is obliged to analyze such a two-vehicle collision on the basis of an actual road accident, and the only information available is the deformation of his own vehicle, it is possible with the aid of the deformation characteristic of his vehicle, if known, to extrapolate via the absorbed energy in such a way as to arrive via the equation.

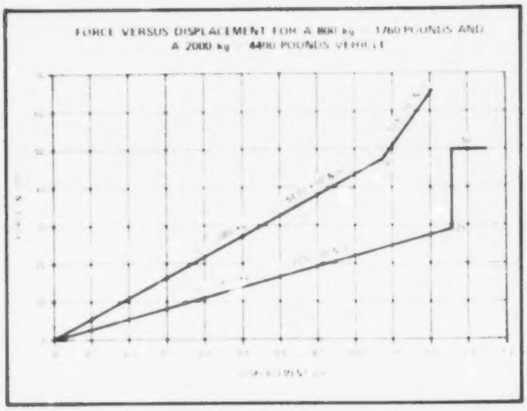

Figure 4

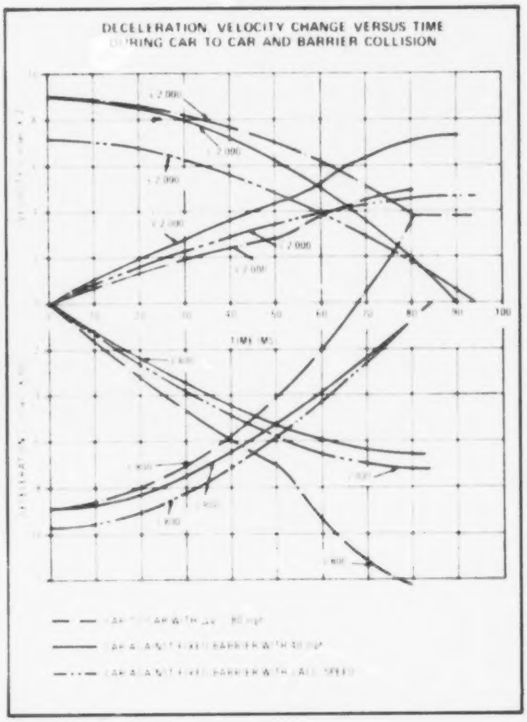

Figure 5

$$\frac{m \cdot v^2}{2} = \int_0^s F \cdot ds$$

the speed on impact. If this calculation is performed in respect of the theoretical case quoted here, we obtain an impact speed for the small vehicle of 44 mph, and a barrier impact speed for the large vehicle of 32.1 mph. In other words, if we use only deformation of the vehicle to define an equivalent barrier impact speed, the results differ markedly from the actual two-vehicle collision. The magnitude of the deviation can be seen from the curves identified in Figure 5.

The problems described here are, in turn, reflected in accident investigations, which are, in fact, intended to provide information leading to the determination of the equivalent barrier impact speed. If these equivalent barrier impact speeds are merely derived from the deformation distance of a vehicle involved in an accident, two errors will result:

1. The actual initial speed and the acceleration/time pattern could be quite different;
2. The relationship between seriously injured and dead persons and an equivalent barrier impact speed is based towards the vehicle investigated.

This means, for example, that when a small vehicle collides with a large vehicle, the equivalent barrier impact speed is higher for the same percentage degree of injury than it would be if only the large vehicle were considered. Despite this, degree of injury and fatal accident figures are frequently used as a function of the equivalent barrier impact speed. This practice occurs, for instance, in the commentary on Standard 208 "Occupant Protection." In another source, (5), this type of plotting procedure is used. If one studies these results more closely (see Figure 6), it is evident that even a small variation in the equivalent barrier impact speed has a considerable influence on the distribution of seriously injured and dead. In the graphs published by the NHTSA, for example, certain personal investigations and also those made by Mackay (6) have been entered. As can be seen, a relatively big deviation exists at higher impact speeds.

In practice, both the value of the equivalent barrier impact speed and the nature of the test method greatly influence the conception of the vehicle, if both are used to define the performance of a vehicle. For this reason the following considerations, related to the frontal collision, must be borne in mind:

1. A new test method must be developed in connection with an equivalent accident speed.

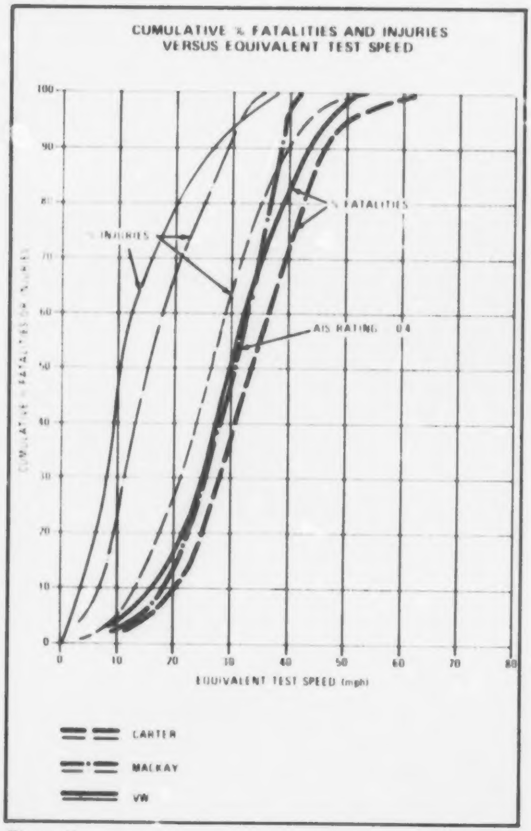

Figure 6

2. When determining the equivalent barrier impact speed, the second vehicle in the collision and its deformation characteristic must be taken into account.

We think that the new ESV-Specification, in respect to front on collision against the five booges, is not sufficient to solve the problem. One possible method of investigating the compatibility of the vehicles would be to measure the impact force exerted by the colliding vehicle on the barrier. With the aid of this information (force-distance pattern), the two-vehicle collision would have to be investigated by computing methods. Whether a simple simulation model in one plane, based on a spring mass system, is adequate for the purpose is open to doubt. It is probable that a spatial simulation model will have to be found.

A limitation in impact force alone would not appear to be sufficient to ensure compatibility among vehicles. Above all, we must remember that deceleration in the passenger compartment cannot reach an unlimited value during such a test. A deceleration level of 35 g would appear to represent an ultimate value, even for considerably improved restraint systems. Another simulation possibility would be actual vehicle-to-vehicle collisions. However, the multiplicity of possible combinations renders this impracticable. A substitute would be to allow the test vehicle to move into head-on contact with a barrier which itself was in motion. The front part of the barrier would have to be arranged to provide energy absorption, so that it corresponded to a standard vehicle. This would also involve simulating the mass of the engine and gearbox assembly. The vehicle to be tested would then be driven against this simulated standard vehicle, and the test criterion could then be a limitation in deformation of this movable barrier. In addition to this test, a second test would be needed to assess the integrity of the passenger compartment. The restraint system of the vehicle could be tested by a horizontal sled apparatus using a predetermined deceleration time impulse. This deceleration time impulse can be established by "mathematical collisions" between vehicles or by installing crash recorders, if the cost of installing such crash recorders in the vehicles appears reasonable.

Attainment of the objective of compatibility is possible in stages of progress. For example, the following general approach could be made:

1. The deformation characteristic of the vehicles, including the masses involved, must be investigated.
2. The necessary deformation characteristic related to the masses involved for compatibility to be achieved must be calculated.
3. Test methods and performance criteria must be developed.
4. After the measures needed have been established, technical solutions for installation in the vehicles must be studied.

Side And Rear-End Collisions

In the same way, when simulating side collisions, non-yielding test barriers are currently used as an equivalent test method. For example, see the barriers required by Standard 208 and the ECE draft. The remarks made in respect to frontal collisions apply here also, namely that the vast majority of side-crashes are collisions between two vehicles. The problem of different masses also assumes significance for side collisions. In contrast to frontal collision simulation, one factor making simulation of side collisions more difficult is that the possible variations of the genuine vehicle-to-vehicle collision are far greater than even those encountered in frontal collisions.

In the simulation of rear-end collisions, in which, again, non-deforming barriers are predominantly used, the energy absorption of the colliding vehicle plays a decisive part in the magnitude of the load imposed on the vehicle struck.

It may be possible here, too, to arrive at a more sensible requirement and resulting vehicle improvements by way of using a movable and deformable barrier.

Summary

Further development of the automobile, with the aim of minimizing accident injury, is undoubtedly an important task, and one of which vehicle manufacturers must be fully aware. However, the costs which the consumer must bear should not be allowed to rise excessively. It is, therefore, of all the more importance for modifications to the vehicle to be accompanied by maximum benefit in relation to cost. The problem of compatibility between road vehicles is undoubtedly to be placed lower in importance than other engineering safety measures, such as increasing the wearing of seat belts. Nonetheless, when new requirements are formulated, for instance the increase in barrier impact speed foreseen in the context of the ESV program of work and in the schedule of the NHTSA for production vehicles, we must ensure that requirements are not made in such a way that, despite an increase in vehicle cost, the effective benefit to the user remains minimal. In the long term, however, efforts must be made to ensure that the performance of vehicles used on public roads in a collision becomes compatible.

Literature

(1) SAE J 850 *Barrier Collision Tests* (February 1963)

(2) FMVSS 208 *Occupant Protection*

(3) ECE-proposal *Economic Commission for Europe Proposals for Side and Rear Impacts*

(4) NHTSA staff paper "A Characterization of Collisions Resulting Damage and Occupant Injury" Research Institute, Office of Accident Investigation and Data Analysis, August 72

(5) Robert L. Carter, *Passive Protection at 50 Miles Per Hour*

(6) Mackay, "Injuries in Collisions Involving Small Cars in Europe" SAE-Paper No. 730 287

SECTION **FOUR** PART **2**

PROPOSAL FOR METHOD OF ANALYZING COLLISION SPEEDS IN REAL ACCIDENTS

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INTRODUCTION

For all the problems with which an engineer is confronted, he needs means of measurement. Generally speaking, he starts off with a given situation, which he has been able to characterize, and must arrive at a new situation, while taking into account the objectives that have been determined for him. He should also have at his disposal means for measuring so as to know whetner he has attained his aims or not.

In the field of motor vehicle occupant protection, this is precisely the problem. The first question specialists ask themselves when examining solutions is: Where is the starting point? The second, which follows immediately, is to what aims should we tend and at what price?

It is for these reasons that we have shown particular interest in the general statistics and the particular multi-disciplinary investigations, both of which allow the present situation concerning road accidents to be represented as analytically as possible.

To carry out this work, our main concern has been:

- to give our attention to the occupant and not to the vehicle
- to understand what kills and grievously injures occupants
- to consider the after-collision state of the vehicle only as a technical support for analyzing the collision
- to supply ourselves with an aggregate means of judgment, thus allowing the existing or design quality of protection to be characterized

On this last point, the most important objective is to characterize the severity of the collision *for the occupant* and not for the vehicle. This characterization is based on knowing the speed variation of the occupant during the main and most violent phase of the collision. It is, in fact, from this knowledge that the performance to be achieved and the cost of solutions may be defined. The cost/efficiency ratio, thus depends closely on the precision of the analysis and the choice of performance.

DIFFICULTIES OF THE PROBLEM

Analyzing the severity of collisions in real accidents is very complicated for the following reasons:

- the great number of possible cases
- the heterogeneity of the car population
- the heterogeneity proper of vehicles

Further, the problem may be divided into two parts:

- analyzing structural deformation
- analyzing the dynamic stresses applied to occupants

The responses of both these elements to the different parameters — speed, deceleration, forces, masses — are not the same, and very detailed analyses must be carried out according to the different cases. Theoretical work has already been carried out in this direction (see bibliography).

COLLISIONS WITH FIXED AND RIGID OBSTACLES

Collisions with fixed and rigid obstacles having large or small surfaces, may be evaluated with sufficient precision if the vehicle in question, or a very similar type, has well known behavior in collisions with both a fixed barrier, head-on or at an angle, and a post.

In the case of multiple collisions, the analysis is more tricky and it is not always possible to evaluate the influence of a collision on the general strength of the remaining structure.

However, we can assume that in cases of collisions with fixed and rigid barriers, the method for evaluating the violence of the collision, compared with a reference test at a known speed, is a good approximation. This is because, in this case, the speed variation of the occupant with relation to the ground is equal to the relative vehicle-obstacle speed, as the latter is reduced to zero during the initial phase of the collision. This phase is expressed as a relative movement between the vehicle's center of gravity and the

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obstacle in question. The stopping distance of the occupant with relation to the ground is, to a considerable extent, equivalent.

COLLISIONS WITH NON-RIGID, FIXED OR MOVING OBSTACLES

These collisions represent a large percentage of accidents and analyzing the severity of the collision for the occupant is, in this case, much more critical. The preceding method for evaluating is no longer applicable, for there is no longer a direct relationship between the deformation of the vehicle considered and the impact speed in collision with obstacle (see Figures 1, 2, 3 and 4).

The deformation of each vehicle is dependent upon the characteristics of the obstacle:

- dimensions and elasticity, in the case of a fixed but non-rigid obstacle,
- masses, stiffness and architecture in the case of another vehicle.

In collisions between vehicles, knowing the impact speed of each vehicle is not the only parameter of interest, for it would give only very limited statistics. Furthermore, it would be difficult to determine the impact speed as there is practically no difference between two vehicles colliding at V km/h, and a vehicle colliding at 2V km/h with a stationary vehicle.

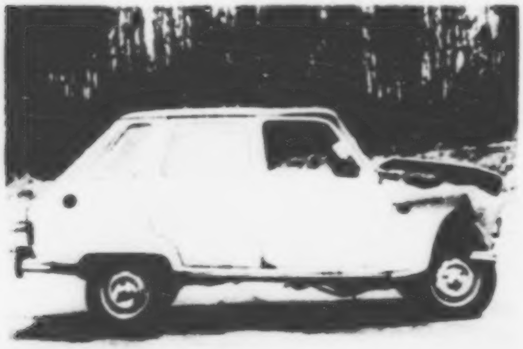

Figure 1

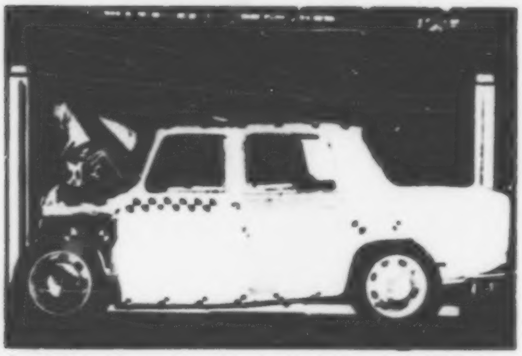

Figure 2

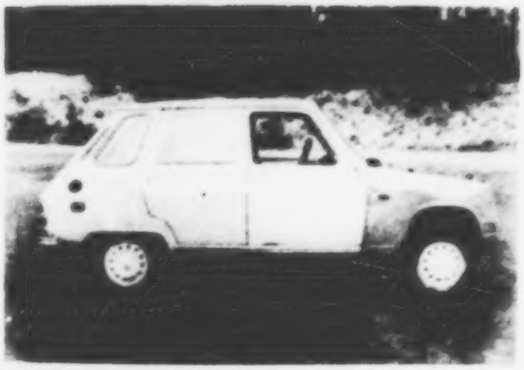

Figure 3

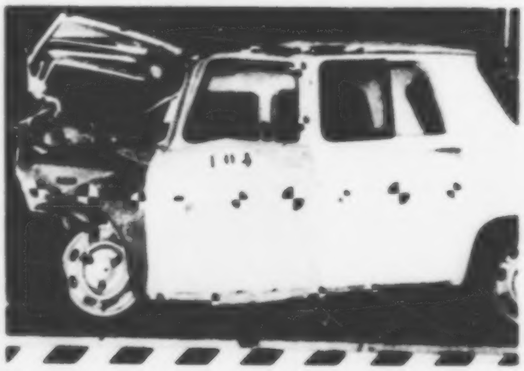

Figure 4

On the other hand, what is important for the results of the collision, with respect to both structure and occupant, is the ***variation of the relative speed*** of both vehicles, V_R, and, more exactly, ***the speed variation*** ΔV of each vehicle. This means that for a given accident between two vehicles, the force will not have been the same for each of them if they differ greatly, and thus will not rank in the same position in the severity scale.

BASES FOR METHOD

After a collision, what do we have at our disposal for analyzing the severity of impact? The two essential elements are the vehicle and the obstacle collided with, which may be another vehicle.

The method for comparing, using a reference collision with a fixed and rigid obstacle, is a general one which allows the overall deformation of the vehicle to be evaluted. This did not seem to us a good method for the cases in which the obstacle collided with is not rigid, particularly when it is another vehicle.

The elements at our disposal for this case are:

- deformation of the vehicle, mainly in such important areas as: side members, chassis, backward movement of engine and wheels (on the other hand, deformation of the wings and bonnet, although sometimes spectacular, has no great meaning where the force of the impact is concerned).
- deformation of the obstacle; the same as above if it is another vehicle, or that occurring in a collision with a rail, a non-rigid post, or a small tree . . .

Analyzing deformation, though it be as precise as possible, presents an interest only if it reveals the forces generated at the interface of the collision as well as the displacement of the centers of gravity, particularly for two vehicles. This problem is made difficult because vehicles rarely collide head-on and because there are great differences in structure. This leads to similar vehicles having a wide range of extremely different crushing forces.

An analytical study of the crushing forces is almost impossible for it is very difficult to determine the influence of the connecting forces between the structural elements. Thus, for example, if a front unit is divided into six parts, three on top and three at the bottom, a certain stress value will be obtained for each individual part when crushing the whole of the front structure against a flat surface; another stress value for the individual part if it is crushed with the adjacent part, and so on.

Nonetheless, in cases where a precise indication can be made of the forces that brought about the deformation, it is quite possible to obtain an indication of the closing speed of the vehicles for a collision with little rotation and no breaking away, by a simple relationship as follows:

$$V_R^2 = \frac{2FX(M_1 + M_2)}{M_1 M_2} \qquad (1)$$

where

V_R = closing speed m/s

F = average stress at interface N

X = total displacement of both centers of gravity towards each other (may be found from deformation) M M_1 and M_2 = masses of both vehicles

All parameters are considered to be known.

The distribution for both vehicles then depends entirely upon the masses:

$$\Delta V_1 = \frac{M_2}{M_1 + M_2} V_R \qquad \Delta V_2 = \frac{M_1}{M_1 + M_2} V_R \qquad (2)$$

$$\frac{\Delta V_1}{\Delta V_2} = \frac{M_2}{M_1}$$

From this information, the initial conditions applied to the occupants ($\Delta V, \Delta X$ thus γ average) for studying devices which reduce injury, may then be reproduced, for example, on a catapult.

On the other hand, it is not possible to put into equation form the structural deformation that is dependent on the types of vehicle encountered. For example, the differences in room left for the occupant inside the passenger compartment are dependent only upon the respective crushing stresses of both vehicles.

DESCRIPTION OF METHOD

The method we are describing is derived from all the preceding remarks and, as its essential characteristic, it uses the vehicle deformation-stress law. This is an easy procedure for all cases of simple collisions and particularly for frontal collisions. It should allow more precise results to be obtained than with the general method of comparison.

The stress-deformation law is obtained from a frontal collision test with a fixed barrier using a dynamometrical buffer that we have already presented. We would remind you that we use from three to six blocks of wood, each fitted with four piezoelectric cells, which allow the stresses in three or six areas to be measured; thus, we obtain all the required stress components, including the general resultant.

Theory Of Method

The vehicles are replaced by one single mass M, and a stiff section whose characteristic is the same as the stress-crushing law for the vehicle (Figure 5) i.e., $F_A = f(x_A)$ for vehicle A, and $F_B = f(x_B)$ for vehicle B.

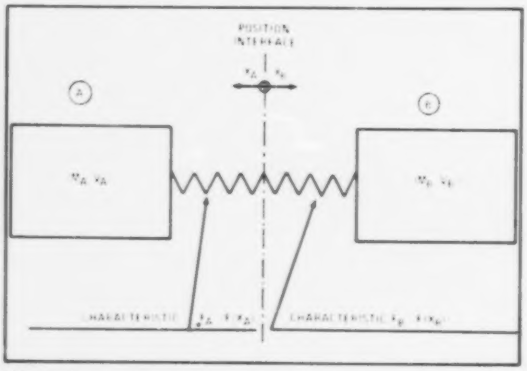

Figure 5

If VA is the speed of vehicle **A** having a mass M_A and VB is the speed of vehicle **B** having a mass M_B, the variation in kinetic energy of the system may be written:

$$\Delta W = \tfrac{1}{2} M_A V_A^2 + \tfrac{1}{2} M_B V_B^2 - \tfrac{1}{2}(M_A + M_B) V_f^2$$

$$V_f = \text{final speed of unit} = \frac{M_A V_A - M_B V_B}{M_A + M_B}$$

This expression allows us to state that the kinetic energy dissipated during a collision of two vehicles will be:

$$W = \tfrac{1}{2}(M_A V_A^2 + M_B V_B^2)$$

$$\tfrac{1}{2}(M_A + M_B)\left(\frac{M_A V_A - M_B V_B}{M_A + M_A}\right)^2$$

which gives:

$$W = \tfrac{1}{2}\frac{M_A M_B}{M_A + M_B}(V_A + V_B)^2$$

in which expression $(V_A + V_B)$ is the relative speed of the two vehicles, i.e. V_R.

On this point, it is to be noted that the total energy dissipated during the collision of two vehicles at V_R is equal to, or less than, the sum of the energy dissipated by each vehicle in a collision with a fixed barrier at $\frac{V_R}{2}$.

Furthermore, we can state that at every moment during the collision: F_A (stress exerted on vehicle **A**) = F_B (stress exerted on vehicle **B**).

The stress, called *interface stress* F_{AB} deforms vehicle **A** (deformation x_A) and vehicle **B** (deformation x_B). The energy dissipated and the total work force F_{AB} may thus be written giving the following relationship:

$$V_R^2 = \frac{2F_{AB} X_M (M_A + M_B)}{M_A M_B} \qquad (3)$$

where

V_R = relative speed of vehicles in m/sec at moment of collision.
F_{AB} = average stress at interface in N.
X_M = closing together of center of gravity of vehicles during the collision in meters ($X_M = x_A + x_B$)
M_B and M_A = vehicle masses in kg.

Thus it can be seen that to apply this relationship, only the following has to be known:

- vehicle masses
- vehicle positions at moment the centers of gravity cease closing together, which may be found by analyzing the deformation of *both vehicles* implicated
- the curve of stresses at interface.

Evaluating Stresses At Interface F_{AB}

To evaluate the product $F_{AB}\ X_M$ used in the relationship (3) the area under the curve $F_{AB} = F(x_A, x_B)$

Thus,

$$F_{AB} = \frac{1}{(x_A + x_B)} \int_0^{(x_A + x_B)} F(x_A, x_B)\, d(x_A + x_B)$$

We know that $F(x_A, x_B)$ in the interval $(0, x_A + x_B)$ is identical to $F_A\ (x_A)$ and $F_B\ (x_B)$ because whichever vehicle is considered, **A** or **B**, the evolution of the stresses is identical, and the same values are gone through.

Let's use, for example, the stress curve $F_A = F(x_A)$

$$F = \frac{1}{x_A + x_B} \int_0^{x_A} F(x_A)\, d(x_A + x_B) =$$

$$\frac{1}{x_A + x_B} \int_0^{x_A} F(x_A)\, dx_A \left(1 + \frac{dx_B}{dx_A}\right)$$

The evolution of x_B as a function of x_A, which would give us the term $(1 + \frac{dx_B}{dx_A})$, is not known, and thus it is necessary to suppose that

$$F_{AB} = \frac{1}{x_A + x_B}\left(1 + \frac{x_B}{x_A}\right)_{\text{mean}} \int_0^{x_A} F(x_A)\, dx_B.$$

After the collision, the only values we have at our disposal are the maximum deformations x_A and x_B, which leads us to suppose that:

$$\left(1 + \frac{x_B}{x_A}\right)_{\text{mean}} = 1 + \frac{x_B}{x_A}$$

which may be written

$$F_{AB} = \frac{1}{x_A} \int_0^{x_A} F(x_A)\,dx_A = \frac{1}{x_B} \int_0^{x_B} F(x_B)\,dx_B$$

When calculating the mean, only the equilibrium of the stresses at the interface is modified (dx_A is used in place of $dx_A + dx_B$). This leads to F_A or F_B being used instead of F_{AB}. We shall see the consequences of this approximation further on (Figure 6).

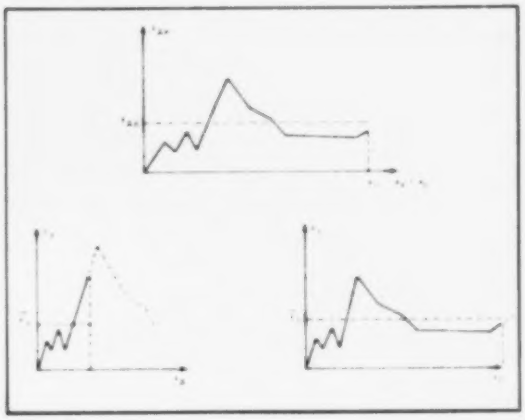

Figure 6

From a practical point of view, this means considering the calculation to be erroneous during the deformation phase, but this error becomes very small at the end of the collision. From another point of view, this also means that the method may be applied even when only one law of stress-deformation for both vehicles is known.

Using The Stress-Crushing Characteristic

The method for obtaining the stress-crushing characteristic by using frontal collision tests with a fixed barrier fitted with a dynamometrical buffer presents the problem of the test speed and the influence of the test speed on the characteristic required. Figures 7 and 8 (collisions of Renault 6 and Renault 12 at 48 and 65 km/hr. with a fixed barrier) show that varying the impact speed leads to a considerable modification of the stress-deformation characteristic.

This modification is due to a small extent to the change of stress brought about by the passenger compartment stopping on the crushed structure at the front (increase from 16 to 19 t); but, for the main part, to the motor drive unit which produces a stress of 34 t. on the buffer at 48 km/hr. and 72t. at 65 km/hr. for the R.12.

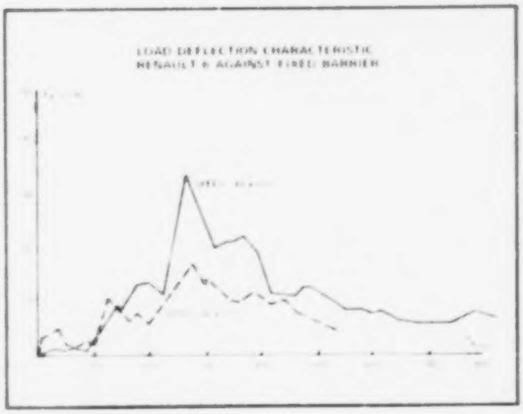

Figure 7

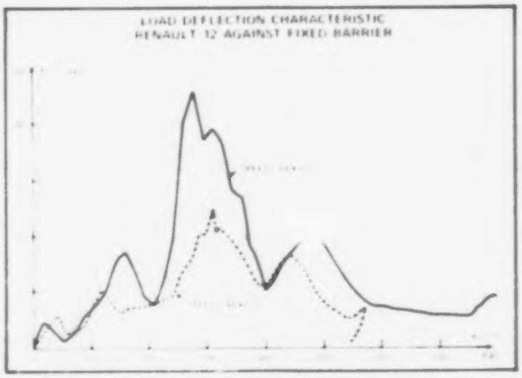

Figure 8

This stress depends for the main part on the speed and the stiff section between the engine and the fixed barrier. Thus, it is certain that this value will be different in the case of a collision with an obstacle having a stiffness different from that for the test with a fixed barrier.

Decomposing the vehicle into, for example, two different parts (on the one hand the passenger compartment, and on the other, the drive unit) would allow us to overcome this obstacle which is one of the inconveniences of this method. A study has been undertaken to account for the influence of the test speed and the impact speed without complicating the analyzing procedure.

The relationship $\frac{1}{2}$ m V^2 = F dx is valid for only one, single, invariable mass. This is not the case for a vehicle comprising several masses which do not all move in the same way during the collision period. The stress F recorded on the dynamometrical buffer must be decomposed into a certain number of elementary stresses F_i whose action is used for stopping the different masses with different movements.

An example of such a decomposition is given for the Renault 12 (Figures 9 and 10) which shows that using two main masses (passenger compartment and engine) leads to a more accurate approximation of the phenomena. But this means prohibitively complicating the method that we wanted to be simple.

The conclusion to be drawn from these remarks is that not just any stress law, at any speed, should be used but a law specified at a speed within a range that is not too far from the collision speed. We shall see further on the differences brought about by not conforming to these remarks, but it should also be noted that the method may be used, if necessary, by recurrence.

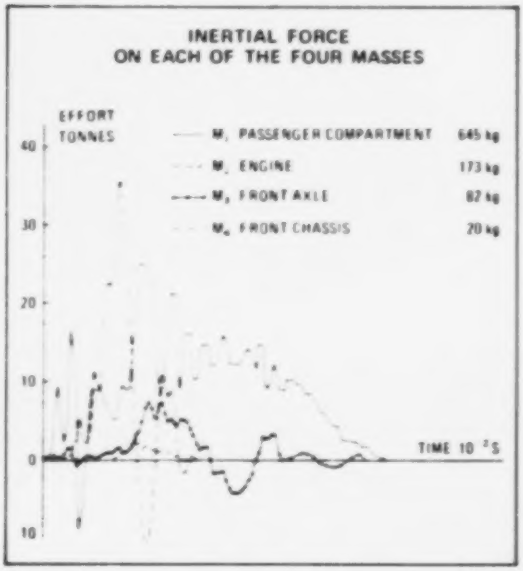

Figure 9

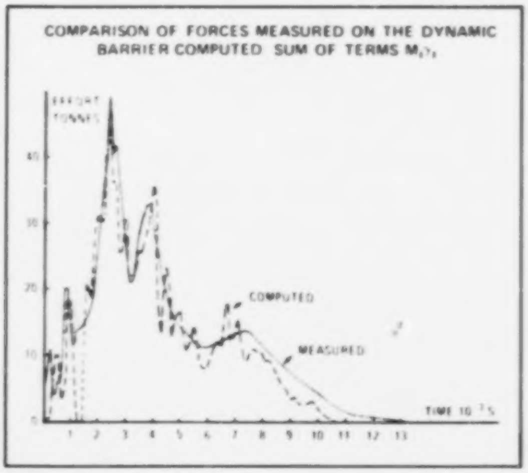

Figure 10

Severity Of The Collision For The Occupant

We have already seen, after having determined V_R, that it is easy to determine ΔV_A and ΔV_B which depend only on the relationship between the masses in presence (relationship 2).

On the other hand, it will be more critical to define *one* magnitude representative in every case of the "severity of the collision for the occupant."

Two cases may be distinguished:

1. If the integrity of the passenger compartment of both vehicles is maintained after the collision, or if the deformation of each vehicle is such that the space required for slowing down the occupant(s) is not prohibitively reduced, it will be possible to define a basis of comparison for the severity of the collision for the occupants of vehicle A with relation to the occupants of vehicle B.
2. If either of the vehicles has been greatly deformed, it is obvious that an important reduction of occupant slowing-down space will be more predominant as far as the effects of the collision on this occupant is concerned.

Irrespective of the method used for analyzing the collision speed, the method used for rating conditions the conclusions. In fact, the method used in estimating a vehicle individually, by referring to a collision with a fixed obstacle, can lead to rating the occupant severity incorrectly.

For example, if we refer to Figures 1, 2, 3 and 4, it may easily be seen that the rating corresponds to, respectively, 30-35 km/h for the R.6 and 60-65 km/h for the R.10 using the method of equivalence with a fixed barrier. If we rely on the respective Δ v for both vehicles, they are rated at 42 km/h for the R.6 and 54.5 km/h for the R.10.

Proceeding in this manner for the whole sample, we reduce the standard deviation in distributing collisions, even if the mean Gauss law is not changed. Now, as we know that the area under the Gauss cure is a function of the standard deviation, choosing one or the other method of rating will have an influence on that speed at which the tests must be carried out for protecting a given percentage of occupants.

Examples Of Application Of The Method – Examining The Results

Following are three examples of applications of the method.

The first two are frontal test collisions carried out at the test center of Regie Renault. The one that follows is an example of analysis of real accidents.

Test Collision Renault 6 – Renault 10

Data

Table A				
Vehicle	Designation	Mass	Initial Speed	V_R real
Renault 6	A	895 kg	46 km/hr	92 km/hr
Renault 10	B	745 kg	46 km/hr	

Stress-Crushing Characteristics (see Figures 7 and 11)

Results after collision: see Figure 12 and Figures 3 and 4.

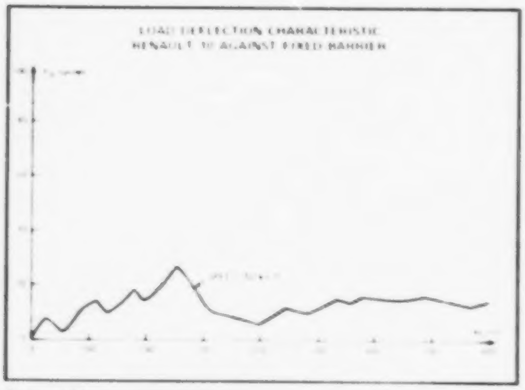

Figure 11

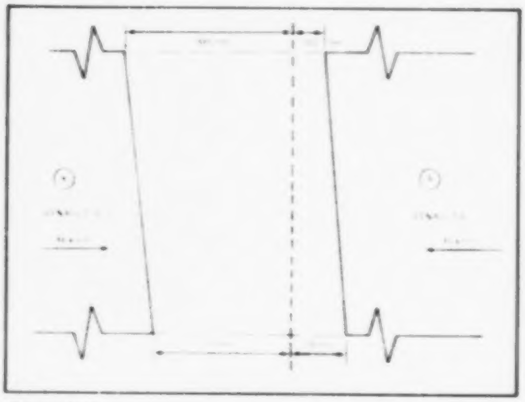

Figure 12

Calculations

Deformation on the left and right hand sides is different for each of the vehicles. We have taken the mean values for the R.6 and the R.10, which also gives the mean value for the closing of the centers of gravity (xM) after colliding.

As we have the stress-deformation characteristics for the R.6 and R.10, we may use either one for determining F_{AB}. This gives us two calculated cases summarized in the following table:

Table B						
[illegible] characteristic used	Vehicle deformation (m)	Value of F_{AB} (N)	Value of x_M (M)	V_R (m/s)	V_R km/h	
Renault 6 (fig. 7) 46 km/h	0.222	105 000	1.0225	23	83	(1)
Renault 10 (fig. 11) 65 km/h	0.880	130 000	1.0225	25.6	92	(2)

As we also have the 65 km/h characteristic of the R.6, we can use this law for determining V_R^2. This gives us one supplementary case of calculation.

Table C						
[illegible] characteristic used	Vehicle deformation (m)	Value of F_{AB} (N)	Value of x_M (M)	V_R (m/s)	V_R km/h	
Renault 6 (fig. 7) 65 km/h	0.222	116 000	1.0225	24.2	87.1	(3)

Discussion On Results

Result (2) is excellent and gives the exact value of the relative speed for both vehicles at moment of collision. Results (1) and (3) are, on the contrary, fairly wide of the mark. This is due to the F_{AB} value being too small in both cases.

By using a deformation of 222 mm for the R.6, we don't take account of the high stress-peak (see Figure 7) that occurs at xA = 230 mm (the mean stress is still not affected, as can be seen from the graph in Figure 19). By superimposing the R.6 and R.10 stress curves, it is concluded that the R.10, having been deformed by more than 700 mm, must have passed through a peak stress of 27 tons; this stress having also acted on the R.6. The latter was crushed by at least 25 cm, the value of xA, for which the value of 27 t was reached.

From this reasoning, the conclusions are drawn that:

1. If both vehicles stop before reaching, or in the middle of a large stress peak, a small error of the crushing distance leads to a gross approximation of the F_{AB} value.
2. Where one of the vehicles is much more deformed than the other, it is preferable to use the stress curve for estimating F_{AB}. In the case studied above, for example, changing the crushing distance from 720 to 820 mm for the R.10 changes F_{AB} by only 0.1 tons, whereas changing the crushing distance from 160 to 285 mm for the R.6 changes F_{AB} from 9.3 to 14.65 tons.

Test Collision Of Renault 6 – Renault 12

Data

Table D				
Vehicle	Designation	Mass	Initial Speed	V_{sc} real
Renault 6	A	940 kg	51 km/hr	102 km/hr
Renault 12	C	1066 kg	51 km/hr	

Stress-Crushing Characteristic – See Figures 7 and 8.

After collision results: See Figures 13, 14 and 15.

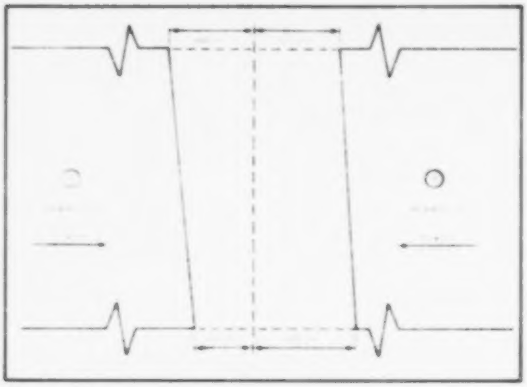

Figure 13

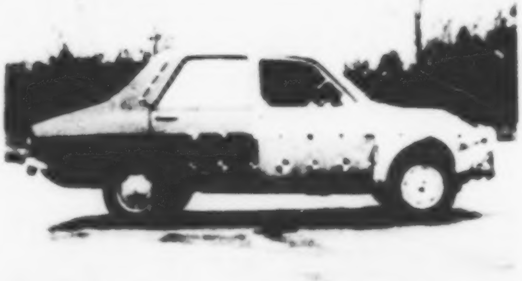

Figure 14

Calculation

The mean closing of the centers of gravity, and the mean deformation of each of the two vehicles were calculated as in the preceding case. Furthermore, we have for each vehicle the two stress-deformation characteristics at our disposal.

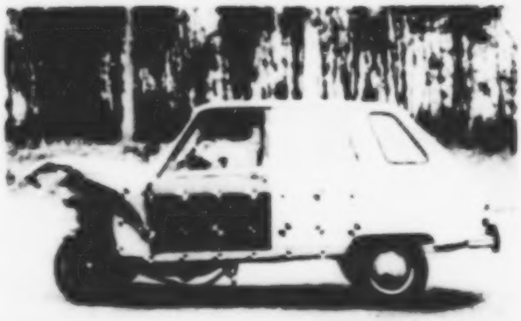

Figure 15

This gives us four cases of calculation which are summarized in the following table.

Table E						
F(x) Characteristic used	Vehicle deformation (m)	Value of F_{AB} (N)	Value of x_{AB} (M)	V_{sc} m/sec	V_{sc} km/h	
Renault 6 Fig. 7 36 km/h	0.584	160 000	0.948	24.6	88.7	(1)
Renault 12 Fig. 8 36 km/h	0.364	206 000	0.948	28	101	(2)
Renault 6 Fig. 7 65 km/h	0.584	252 000	0.948	30.9	111	(3)
Renault 12 Fig. 8 65 km/h	0.364	354 000	0.948	36.7	132	(4)

Discussion Of Results And Conclusions To Be Drawn From The Two Examples

For this example again, some of the results are aberrant.

Result 1 for having used too small a F_{AB};

Result 4 for having used to great a F_{AB}.

The large difference obtained in the last calculation results from taking into account the R.12 engine peak stress at 65 km/h. The simplicity of the model used for representing vehicles does not allow the engine mass at the front to be taken into account. Its behavior is quite different in a collision with a buffer (determining the stress-deformation characteristic) and in a vehicle/vehicle collision. It is only at the price of this inconvenience that the simplicity of the method may be maintained.

However, in spite of the lack of precision of this procedure, it is interesting to note that it avoids the gross errors brought about by using the method of comparing vehicles having undergone a vehicle/vehicle collision with those having undergone a vehicle/wall collision.

Comparing both R.6's, one for the R.10 collision and the other for the R.12 collision is sufficiently eloquent from this point of view. It may be noted that the mean results obtained for the last example give a value of 108 km/h for an actual speed of 102 km/hr.

Severity Of The Collision For The Occupants

Each test vehicle was fitted with a dummy restrained by a 3-point belt in the front passenger seat.

If we calculate the ΔV for each vehicle, we obtain:

$$\Delta V_A = \frac{102 \times 1066}{2006} = 54 \text{ km/hr}$$

$$\Delta V_C = \frac{102 \times 940}{2006} = 48 \text{ km/hr}$$

But, as the survival space has been maintained, we may also compare the results obtained on the dummies.

Table F

	S I	HIC	Thorax maxi	S I thorax
Passenger Vehicle A	1607	1145	45	312
Passenger Vehicle C	1165	943	45	290

Second Application To A Real Accident. Collision Renault 6/Estafette

Data

Table G

Vehicle	Designation	Mass	
Renault 6	A	930 kg	V_{ij} to be determined
Estafette	B	1250 kg	

Stress-Crushing Characteristic (See Figure 7)

We have only the R.6 curve at our disposal.

Results Of The Collision: (See Figures 16, 17, and 18)

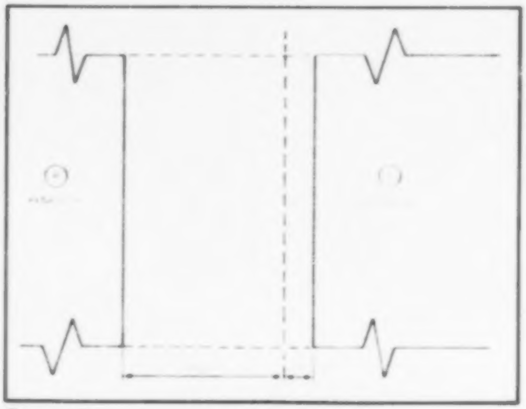

Figure 16

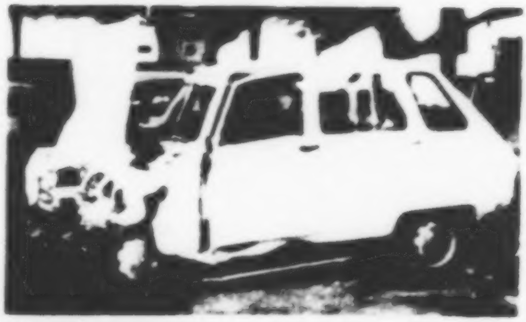

Figure 17

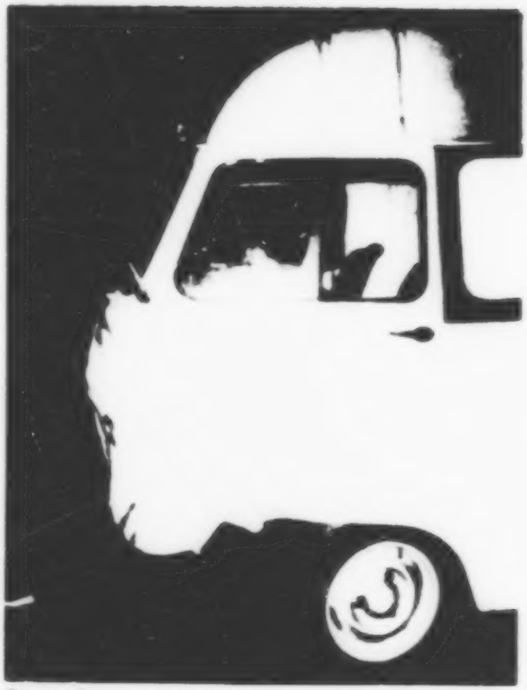

Figure 18

Calculations And Results

Table H					
F (x) characteristic used	Vehicle deformation (mm)	Value of F_{mm} (N)	Value of x_{sp} (M)	V_H m/sec	V_H km/h
Renault 6 65 km/hr	800	220 000	0.900	27.2	98

$$\Delta V_A = 56 \text{ km/hr}$$

$$\Delta V_B = 42 \text{ km/hr}$$

The driver of vehicle A, who was not belted in, suffered a head injury with loss of consciousness lasting longer than fifteen minutes, contusions to the face, thorax and knees. He recommenced active life after eight days in hospital.

The driver of vehicle B, who was retained by a 3-point belt of present day fabrication, suffered a head injury with loss of consciousness lasting less than 15 minutes, contusions to the face, thorax and knees. He recommenced active life after thirteen days in hospital.

CONCLUSIONS

Presenting a method of evaluation of collision speed in real accidents as objectively as possible proves that there is no simple, unquestionable solution, on the theoretical side, which entirely covers the complexity of reality in road accidents.

However, we think we are making a much smaller error when using a method based on the stress-deformation laws for each vehicle, instead of the one generally accepted which is based on comparing the deformation of the vehicle involved in the accident with the same deformation obtained in a collision with a fixed and rigid obstacle. The estimated speeds in this last case may have no relationship, depending on the type of real obstacle encountered, with actual ones.

Applying this method to actual cases has led to a classification of collision severities that is appreciably different from what it would have been by applying the comparison method.

This application shows that the passenger compartment stopping distances with relation to the ground are greater in real accidents than to what they would have been in a collision with a fixed barrier, thus changing the "severity" of the collision for the occupant.

Note On The Practical Use Of The Method

The stress-deformation laws for the vehicles may be represented in the form of graphs showing the average stress as a function of X (Figures 19, 20 and 21).

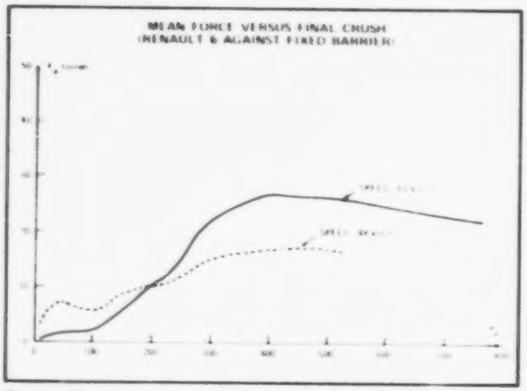

Figure 19

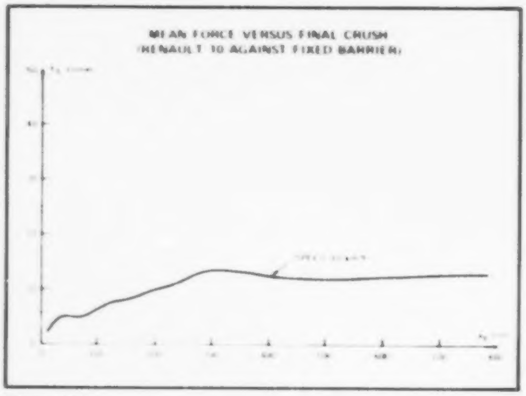

Figure 20

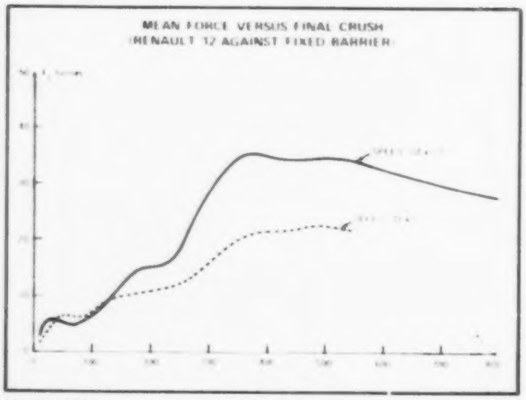

Figure 21

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CRASHWORTHINESS SEMINAR

SECTION FOUR PART 2

THE EFFICIENCY OF BODY ENERGY ABSORPTION AND PASSENGER PROTECTION DEVICES

TERUO MAEDA, *Manager*
Safety Vehicle Project
Safety Research Department
Nissan Motor Company, Ltd.

As was explained the other day, the values used to meet the specified requirements for the Nissan ESV were determined by adopting two built-in devices. In order to select these devices, experiments and comparative evaluations were made on a trial body with a built-in sub-frame and a large size hydraulic damper. Trials and evaluations were also made on a built-in frame type vehicle, the design of which was prepared by the Cornell Aeronautical Laboratory at our request. The Laboratory's results demonstrated that E1 and E2 body types are desirable in smaller size vehicles.

In order to develop the E1 body, a special monocoque body was designed and built for each trial car. A systematic analytic study was then made on every vehicle. The basic idea behind this procedure was to utilize the monocoque body to the greatest extent possible to determine the energy absorption of the vehicle. It was found that during an 80 km/h collision, about 50% of the energy was absorbed by the front part of the dash board. The balance was absorbed by the damaged floor, the remainder of the dash board, the power train and so forth, within the enclosed car. Consequently, the body weight of the car was increased as little as possible.

However, although the body deceleration curve continued to peak during the latter half of the crash, it was possible to minimize the risk to the passengers by using an air bag to absorb some of the shock. Needless to say, the air bag must be put into operation quickly, even if this results in a decrease in its reliability.

The E2 vehicle has a broaching unit with an extremely high energy efficiency. The object in this case was to achieve close to 90% energy absorption in the front part of the dashboard. Owing to the special quality of the broach, a relatively flat body deceleration curve was obtained. Combined with a seat belt, this made it possible to measure passenger safety. In this instance, it was necessary to install a preloading device for the belt and an autoretractable steering unit.

As a result of using two systems, we were able to develop a broad base for use in constructing an ESV car. However, as previously noted, the two experimental vehicles have a number of unsolved problems which made them impractical for production at the present time. Hopefully, we will be able to utilize some of the specific features of each experimental vehicle in future designs. It should be noted, however, that the nature of these future designs will be dependent on the amount of money available to build them. It will be necessary to determine how much money ought to be spent for the exterior of a vehicle and how much for the interior in terms of passenger safety.

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CRASHWORTHINESS SEMINAR

SECTION FOUR

PART 2

REMARKS ON THE TRANSPORT AND ROAD RESEARCH LABORATORY

R. DUNCAN LISTER, *Head of Vehicles Division*
Safety Department
Transport and Road Research Laboratory

My remarks are of a more general nature than the ones you have just heard; nevertheless, I hope they are relevant. First of all I will give you a brief explanation of where the Transport and Road Research Laboratory fits into this theme, because it may not be quite familiar to some of you.

The Transport and Road Research Laboratory of the United Kingdom is a government research establishment entirely funded by the government and therefore independent of the industry. But we have no legislative authority. We only advise.

The laboratory provides technical and scientific advice and information to help in the formulation, development and implementation of government policies relating to roads and transport, including their interaction with urban and regional planning. So you can see the scope is very wide.

Now, included in that, and this is where I am particularly concerned, is the study of road accidents and methods of reducing their frequency and severity. So this is where we fit into the scheme and why you perhaps hear me from time to time trying to broaden the scheme, broaden the aspects of reducing accidents and injuries.

The laboratory has about 1,000 staff at the moment, and about half of those are qualified in some discipline, engineers, physicists, psychologists, mathematicians, medical people. I think if you name a discipline you will find it somewhere in the laboratory. In addition, our truck facilities and dynamic test facilities are quite wide. Well, that is just a brief explanation.

With regard to what Dr. Mannella said this morning about NHTSA's intermediate specification being really its goal, is similar to what we submitted as one of the papers in our official papers. We have not called it a specification for the very same reason. These are our requirements for what we have called the practical safety car, which we believe would be achievable and would provide some improvement in the accident and injury work.

Now, to indicate how we arrive at our priorities and so forth, we make great use of accident and injury data. This is data that we have ourselves. You may not be aware of it, but in the United Kingdom all injury-producing accidents are reported, and there is a very detailed system for putting these on computers and then analyzing them. The more detailed aspects, of course, we have to investigate ourselves. And in this kind of work we are also aided by the work that is done at Birmingham University, where Dr. Mackay is the leader of the team, so that we have the detail as well as the general information.

This helps us to guide our priorities and determine where we put our emphasis. It is on the strength of the accident data and statistics that we are pressing more and more for work on the pedestrians; that the front of the vehicle should take much more account of pedestrians. And I believe this would also be the case in Japan. I think the need is probably here, too.

Another use that we make of our injury work, both detailed work and data collecting, is the reproduction in the laboratory by dummy simulation, not computer simulation, but actual physical damage simulation, impact test devices, and so forth, so that we can get a quantitative measure of what is involved in producing these injuries.

In passing, I might add that we also do the same thing for what you call here, the "active safety." We try to reproduce the conditions that generated the accident in the first place.

If I might now turn to just a few general points. We have heard a lot about the different injury criteria; but what really do we mean by this? At the laboratory we take this criteria to be skeletal fracture. That is the point at which we say, that is the critical point, the body cannot stand any more loading. That is fracture generally, or dislocation of the hip, or the joint; and of course there are special conditions for the head.

Now, having got all these data from the injury and simulation work, we relate them to our test devices, whether they are dummies or what have you. As long

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as we know that when these test devices read a certain number or a certain figure, and the injury or skeletal fracture does or does not occur, then we are much closer to relating our work to the real world. And this, we think, is the most important thing.

I would like to relate one example involving the head injury criteria used for assessing head injuries on dummies. Practical tests showed that seat belts would not meet the criteria, and yet we had very clear evidence from the many thousands, hundreds of thousands, in use on our roads that although people had been injured and the loading had been sufficient perhaps to cause some kind of fracture, we were not getting any head injuries. This was a clear case where the head injury criteria as used at that time did not comply with the real world experience.

I am also critical of the femur loads. I can't quite remember the figure here. I think the figure given is about 1,700 pounds. In our accident and simulation work this is too high. We have determined this is from using dummy test devices on the human form. The figure at which fractures or dislocations begin to occur is above a figure of about 4 or 5 kilonewtons, which is much lower. Again, I think this needs to be taken account of.

In addition to using accident statistics and injury data for determining priorities, you need to use the same techniques for monitoring the benefits or lack of benefits that arise from any design changes. And this accident recording should be a continuous process so that one can see what effects these design changes are having.

In the near future we will be bringing out a report on the energy-absorbing steering column.

In a similar way, changing the subject slightly, the form of crash testing and the levels should be related to the types of accidents that are occurring. We have had brief references to that from other speakers with the added proviso that we can use it to encourage any particular line of design development which we have reason to believe would be helpful. In that respect, I am very critical of the earlier pole test. The revised test is a little more reasonable for two reasons: the incidence of the pole, or tree accident, is perhaps more in an urban environment, and these things are considered for any new highways or future highways. In fact, there are usually no poles to strike, but if there are any, they are well protected by guard rails. Therefore, why have a test that is not going to be a frequent occurrence in future accidents? In the urban area, of course, the level of impact is much lower. This is where our highway people come in; it is a highway problem.

Well, I think I have said enough to indicate our general views and our general approach to this work. We seem to be getting much closer together than we were previously. The NHTSA views and ours and other European countries are in fact much closer than they were two years ago.

CRASHWORTHINESS SEMINAR

SECTION FOUR PART 2

COMMENTS ON THE ESV PROGRAM

VITTORIO MONTANARI, *Director for Applied Research*
Department of Motor Cars
Fiat SPA

The technical presentation delivered by Fiat yesterday ended on a hopeful note on future regulations. A quick look at the new 3,000 pound ESV Specification released by DOT shortly before this conference would not seem to justify these hopes. In fact, the specification requirements reflect very little of what has been accomplished in the different countries since the international research program was launched.

The changes in respect to the first 4,000 pound ESV Specification are limited to a modest recognition of the existence of the compatibility problem, and to a modification of some requirements which were marginally important, like low velocity impacts, or were manifestly excessive, like front and rear end collisions against a pole. On the other hand, the severity level of other tests has been raised in spite of the serious difficulties that had already become apparent at the previous severity level: particularly the side impact, both against a pole and in car-to-car collisions. However, we will keep on hoping until September of this year when the final DOT specification for an optimized family sedan should be released.

If the September specification is in line with the present one, we must come to the unfortunate conclusion that the huge amount of work conducted so far, (estimated by Mr. Beggs at up to $100 million, but certainly much higher), all will have been of very little use. What is even more serious, any future work will also be of little benefit.

In such a case, we would be forced to become passive bystanders and would have to accept the imposed standards. There would be no need to seek our collaboration; our mere obedience to laws would suffice.

We think that after two years of work since the start of the common program, the different studies conducted by the industries have already given sufficient information and results to set new goals which take into account not only the technical, but also the economical and social realities.

Briefly, our opinion can be reassumed as follows:

- At the present state-of-the-art it is possible to meet, hopefully and in a relatively near future, the survival space requirements, provided the conventional severity level of the different crash tests is commensurate with the 30 mph impact for the front end collision against barrier. Levels exceeding this speed would lead to technical and economical implications which, in our opinion, are not at all proportionate to the benefits obtainable.

As far as crashworthiness is concerned, we tried to show that the most important problem yet to be solved is compatibility. It is on this problem that the major part of all our work and studies should be focused.

- Another basic objective is the improvement of restraint systems which really give effective protection of occupants. Let us try to reach these goals as soon as possible — we will thus render a great service to mankind. Any more ambitious program would at the present time be irrational and would represent only a waste of time and money.

We really have devoted a considerable amount of effort to the ESV program. But we should not forget that from a cost-efficiency point of view we have responsibilities not only to our management, but also — or even more so — to our customers and the public opinion.

CRASHWORTHINESS SEMINAR

SECTION FOUR PART 2

MOTOR-VEHICLE ACCIDENTS IN JAPAN

YOSHIO SERIZAWA, ***Manager***
Safety Research Department
Nissan Motor Company, Ltd.

1. Motor-vehicle accidents were reduced from 1970. Motor-vehicle deaths and nonfatal injuries were reduced from 1971 (Figure 1).
2. Motor-vehicle nonfatal injuries per 10,000 registered vehicles were reduced from 1969. Between 1966 and 1972, motor-vehicle deaths per 10,000 registered vehicles were reduced 51.7 percent from 14.9 to 7.2 (Figure 2).
3. Death rates in Japan and Europe are different from death rates in the United States. In nations other than the United States, there are problems of deaths, except vehicles (Figures 3, 4, 14).
4. From 1970, injuries per 10,000 registered vehicles were reduced; but police actions increased (Figure 5).
5. Forty-eight percent of fatal accidents were collisions with pedestrians or bicycles (Figure 6).
6. Sixty-eight percent of pedestrian deaths were crossing or running out in roadway (Figure 7).
7. Sixty percent of drivers involved in accidents were under thirty years of age (Figure 8).
8. Sixty-one percent of the driving experience was under five years (Figure 9).
9. Accidents were reduced by improving road equipment (Figures 10 and 11).
10. Eighty-five percent of vehicle speeds in accidents were under fifty kilometers per hour, and ninety-nine percent under eighty kilometers per hour (Figure 12).
11. Fifty-eight percent of the collisions were head-on, and one percent overturning. (Figure 13).
12. Deaths per 10,000 registered vehicles and per 10,000,000 vehicle kilometers of small cars were less than those of large cars (Figure 15).

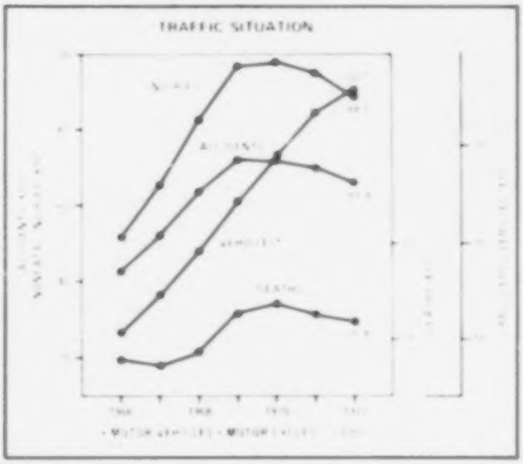

Figure 1

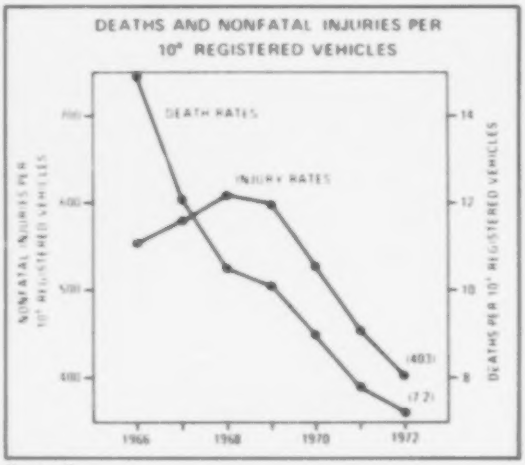

Figure 2

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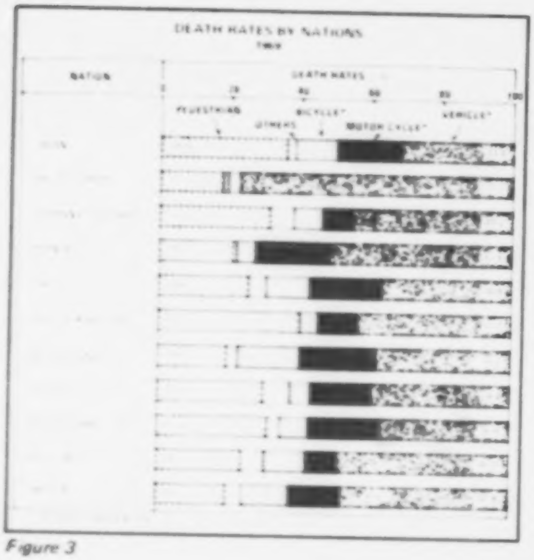


Figure 3

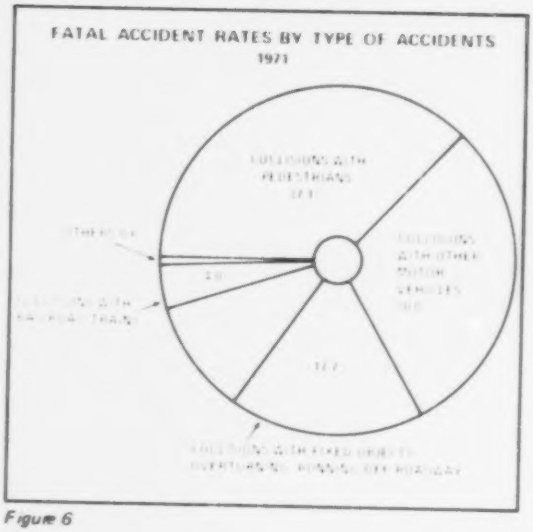


Figure 6

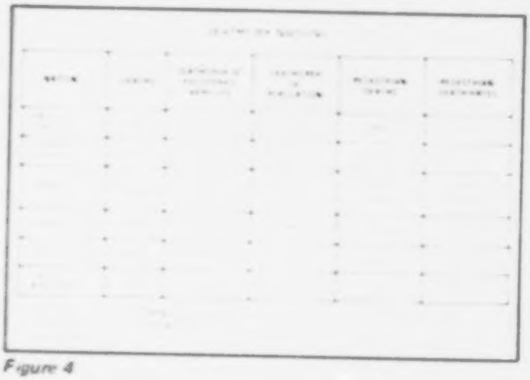
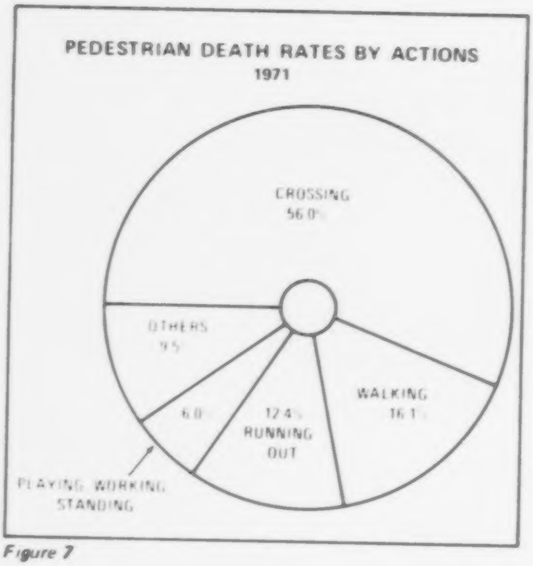

DEATH BY NATIONS

Figure 4



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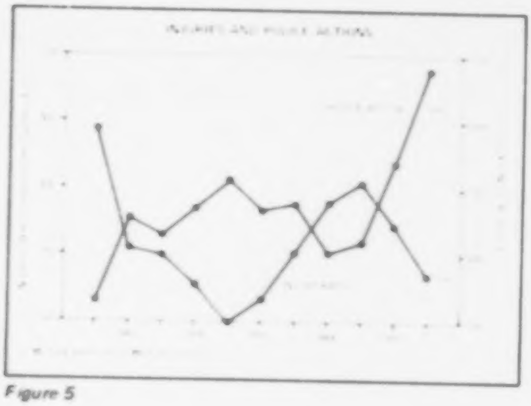

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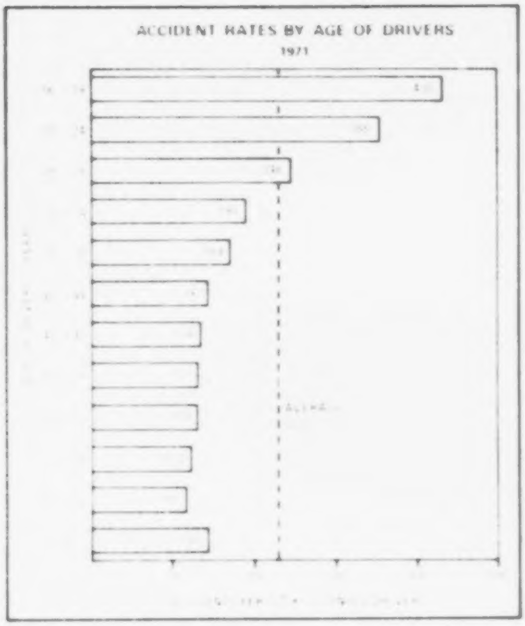
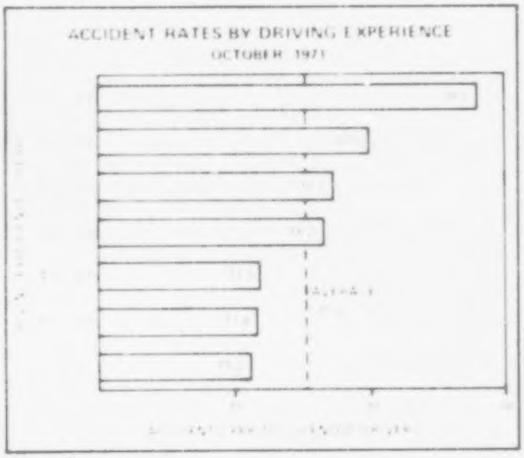


Figure 8



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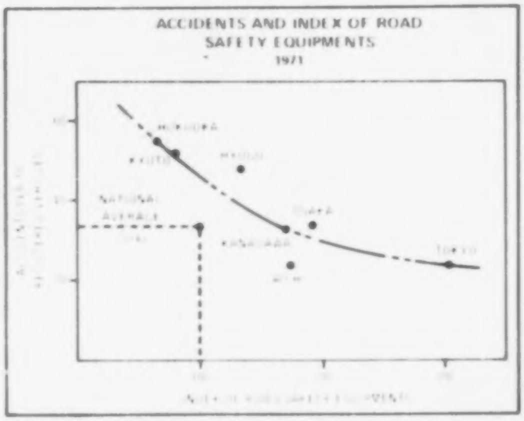


Figure 10

EFFECT TO ACCIDENTS BY IMPROVING ROAD EQUIPMENTS
1971

ROAD EQUIPMENT	ACCIDENT REDUCING RATES	DEATH REDUCING RATES
STOP SIGN	60 %	85 %
NON SLIP ROAD	57	70
PEDESTRIAN BRIDGE	54	78
[illegible] STOP SIGNAL	48	58
SIGN FOR PEDESTRIAN TO CROSS	37	65
LIGHTING	29	56
NO PASSING SIGN	24	58

Figure 11

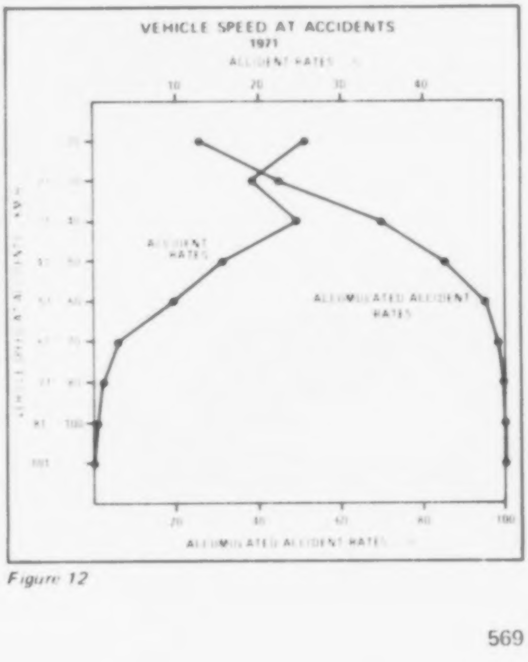


Figure 12

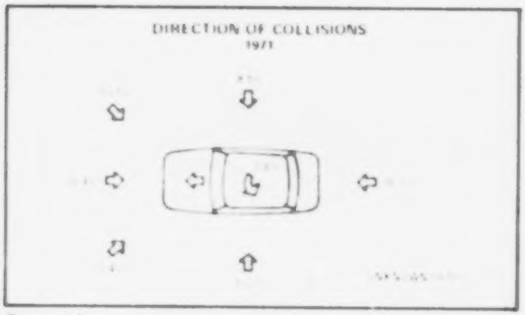
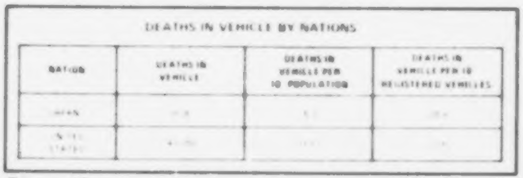

DIRECTION OF COLLISIONS
1971

Figure 13



DEATHS IN VEHICLE BY NATIONS

NATION	DEATHS IN VEHICLE	DEATHS IN VEHICLE PER 10 POPULATION	DEATHS IN VEHICLE PER 10 REGISTERED VEHICLES
JAPAN	[illegible]	[illegible]	[illegible]
UNITED STATES	[illegible]	[illegible]	[illegible]

Figure 14

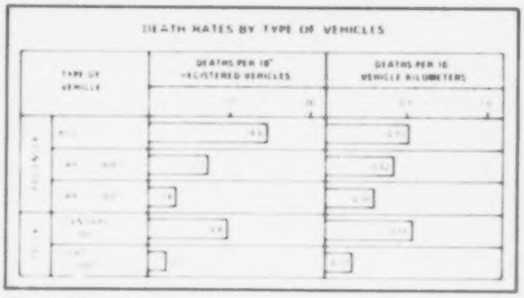


Figure 15

CRASHWORTHINESS SEMINAR

SECTION FOUR PART 2

COMMENTS ON THE 3,000 POUND VEHICLE SPECIFICATION

COLVER R. BRIGGS, *Director*
Automotive Safety Planning and Research Office
Ford Motor Company

I appreciate having this opportunity to comment on selected aspects of the recently circulated 3,000 pound ESV specification. I will start first with some of the accident-avoidance type of considerations, but barely touch upon them, as they are not in the main thrust of the proposed 3,000 pound ESV program, and are not the selected subject for this seminar.

The vehicle controllability requirements proposed for the 3,000 pound ESV are much the same as for the 4,000 pound ESV. Although they were not all successfully achieved in the 4,000 pound prototypes, we anticipate that most of them could be met in second generation vehicles. There are a few details that may not be realistic, however. For example, we are not sure that the stopping distance and brake efficiency requirements are compatible. Some of the new handling tests are going to be difficult to carry out because of complexities of test apparatus and procedure, and because of possible hazard in some of them.

More important, however, if we believe that the controllability requirements can be mainly achieved, we do not necessarily also believe that the vehicle would be safer or that these are the most appropriate requirements — or that they are even safety-related.

The driver visibility specification can also be physically met, but we must remember that its relevance to safety has not been demonstrated. The visibility specification is based on the DOT proposals in the two regulatory docket notices which have been issued on this subject, and the recent report that the docket would be closed suggests that there are widely felt doubts about them. We were pleased to see that the visibility requirements did, however, incorporate some elements drawn from industry recommendations.

We have some reservations about some of the other operational features, such as the over-optimistic goal for high speed vehicle acceleration, but I would like to go on now to consideration of the crash performance goals.

We believe the goal of the 3,000 pound ESV development ought to be mainly that of achieving occupant protection. The limits that the specification places on crash deceleration of the vehicle itself, characterized by a speed-sensitivity curve, allows only an intermediary means for accomplishing occupant protection, and possibly is not the only way of reaching that objective. The combined requirements of the speed-sensitive front structure, as indicated in Figure 6 of the specification, and the "aggressivity" limits, as indicated in Figure 8 and Table 3 of the specification, put considerable constraint on the energy-absorption levels and the maximum force levels. It would seem that the proposed specification is intended to influence the design approach into one using hydraulic struts, such as those used in some of the early safety cars, and in related research projects. While this may seem to some a good approach, others will feel that the complications of hydraulic struts (and their questionable feasibility for anything but direct head-on hits) is not appropriate — especially for a 3,000 pound car.

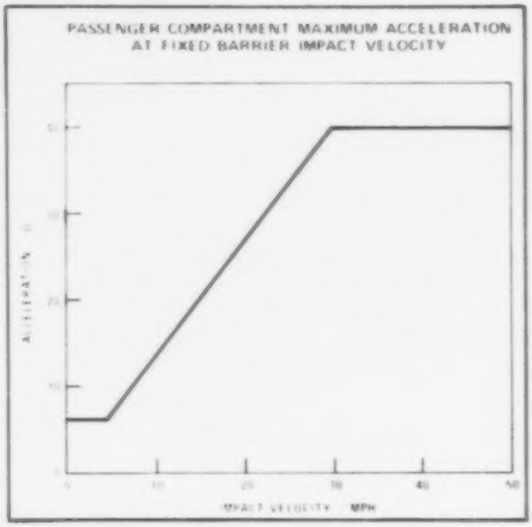


Figure 6 of the Specification

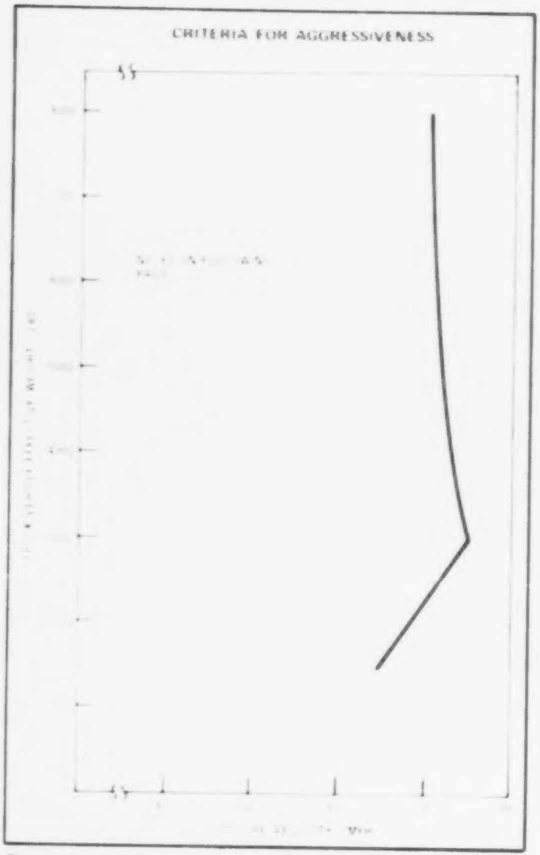


Figure 8 of the Specification

It seems to us that a more acceptable alternative to essentially duplicating one of the previous research solutions — the AMF development — would be to undertake an intensive study project in order to find alternative methods of achieving the desired occupant-protection goals. These goals should be carefully considered so that, for example, "aggressivity" is clearly related to occupant survival, rather than to vehicle damage.

We believe that aggressiveness studies should be oriented toward a more realistic description of the "other" car — the car that would be struck — in terms of a more representative range of both weights and crush properties. Reducing the range of coverage should at least make the program more manageable. We are pleased, however, that the proposed specification for front impact damage resistance has been related to the realistic speed of 5 mph.

The side impact requirements called for in the new set of specifications are, in our judgment, not realistic. Experience to date suggests that the 60 g lateral chest acceleration called for in FMVSS 208 will be difficult enough to meet, which is far different from the 30 g objective called for in the new ESV specification. We believe a realistic requirement would be for 20 mph side impact, and that intrusion should be related to occupant injury — meaning that an occupant zone should be defined limited to areas where intrusion might be relevant. Inasmuch as dummy kinematics are so unpredictable in rollovers, we strongly urge that the roof crush requirement of FMVSS 216 be substituted for the proposed car rollover specification.

Speaking of test dummies, we should not forget that the dummy scene is confused at present, and that it would be inappropriate to set test levels for a new ESV development that did not depend on the test dummy eventually established for measuring occupant protection performance.

With respect to occupant restraints, it is doubtful that any passive restraint known of today could satisfy both the occupant packaging and crash speed requirements proposed for the 3,000 pound ESV. It seems much more likely that a successful design could be achieved within the vehicle size specified if 40 mph were the crashworthiness goal.

The pedestrian and cyclist's protection specification proposed for the new 3,000 pound ESV strikes us as being inappropriate for this particular program. This is not because we believe such protection is not needed, nor because we believe it should not ultimately be included in United States ESV development, but because it is a major new area of study in its own right. That little is known of the United States' situation is acknowledged in the DOT proposal, for the specifications are twice preceded by the expression, "in the absence of technical justification to the contrary . . ." We think this subject should be considered separately from the ESV program.

I would like to remind you of some considerations which those of us who are expected to implement these designs often have before us — those of safety-relatedness. While it is true that the first ESVs did perform reasonably well in the operational tests and in the crash tests — even though they did not fully meet the objectives for which they were designed — this does not mean that those designs are correspondingly safer than the designs for today's production vehicles.

There is a difference between test results obtained under the unique circumstances that were specified for the experimental vehicles and actual safety performance in the real world.

We can, with almost certainty, say that the amount of *safety improvement* was *not* proportional to the test results. Expressed another way, we can say that whatever the ultimate safety improvement, it was not commensurate with the cost implied by the type of designs that were adopted.

The true measure of safety performance of the ESV type of vehicle will have to remain conjectural at this point, because we do not know how to translate these crash tests into real-world terms. Ultimately, we would have to note what happens when we introduce these kinds of cars into traffic. Then we could raise the real question: was there a net gain or loss in fatalities and injuries as a consequence of having these cars on the road? When these cars are on the road they will be involved in accidents which, on the average, are quite different and mainly less severe than those specified for the test. There is no good way of estimating the extent to which these ESVs — which out-performed ordinary sedans in the 50 mph barrier crash test — would show any advantage over the whole range of real-world collisions. Some calculation should be made of the net effect when their probably high aggressivity in side impacts is considered. Some calculation should also be made of the net effect on the ESVs own occupants at lower speeds. Those who are impressed with the need for high speed protection because of the significant numbers of fatalities which occur at speeds above, say, 40 mph, may not be thinking of the enormously greater numbers of people who are *exposed* to collisions at speeds below that level. The numbers of people who are *exposed* increases extremely rapidly for every mph of reduction in barrier equivalent crash speed. We feel the crash test speed for evaluating ESVs should be established at a level where there is a more appropriate balance between significant collision exposure and fatalities. We believe that 50 mph is too high — that 45 mph is too high — that programs targeted on these objectives will not be as productive as ones directed toward a more orderly development of crashworthiness technology.

In summary, I wish to express my hope that those who undertake to design this new ESV venture in its final form will profit from our previous ESV experience — that they will revise the tentative goals, which in many respects seem even more radical than those for past programs, and will make them more realistic. What could not be attained with the 4,000 pound ESV — even with a considerable weight overrun — we believe is unattainable in a 3,000 pound vehicle. By specifying a 3,000 pound vehicle, recognition has obviously been given to the undesirability of a "safety at any cost" posture. It would seem, then, that the research question should be, "how much safety is it possible to derive from a 3,000 pound vehicle?" A study project oriented toward determining the maximal amount of safety achievable within the practical requirements of a 3,000 pound, $3,000 vehicle would make more sense to us and I would hope to you, and the public — than to continuously try for more and more on the assumption that anything is achievable if you just try hard enough. It is always difficult to crowd ten pounds of turnips into a five pound bag.

We believe it would be useful to carry out a study project in which the limits of practical achievability are determined and which estimates the probable real-world safety performance of designs which meet crash test criteria. However, if that kind of a study project does not fit in with the thinking behind the ESV program, and if specific performance goals are to be pursued in an ESV development, we strongly suggest that it would be more realistic to aim for a 40 mph frontal requirement, for correspondingly reduced requirements in other impact modes, with relaxed limitations on vehicle deceleration and a narrower range of aggressivity expectations. We believe that the previous experience with ESVs has demonstrated the need to take an evolutionary, rather than revolutionary approach to this subject.

It is our hope that such a plan will be developed for the next generation of ESVs.

Table III of the Specification

MINIMUM ENERGY ABSORPTION REQUIREMENTS
OF 3000 POUND ESV

Bogey Vehicle Weight* (lbs)	Frontal Head-on Closing Speed (mph)	Required Energy Absorption in 3000 Pound ESV (ft/lbs)
1500	64.8	33,746
3000	74.8	67,491
4500	72.7	212,942
6000	71.3	212,942
8000	70.2	212,942

*Test Bogey vehicles designed with the following characteristics in fixed rigid barrier impacts

Test Bogey Weight (lbs)	Barrier Impact Velocity (mph)	Constant Dynamic Crush Load (lbs)	Dynamic Crush Stroke (inches)
1500	50	45,000	28.4
3000	50	90,000	28.4
4500	26.4	90,000	13.9
6000	25.2	90,000	16.9
8000	23.4	90,000	19.5

CRASHWORTHINESS SEMINAR

SECTION FOUR

PART 2

PEDESTRIAN PROTECTION

DR. MAX DANNER, *Research Scientist*
Association of Third Party
Accident and Motor Traffic Insurers (HUK)

NATURE AND EXTENT OF INJURIES TO PEDESTRIANS

Test Material

The results presented here stem from approximately 100 traffic accidents submitted by the courts and defending counsel for detailed study and examination, so that the data ascertained on

a. impact velocity and speed of the car
b. final position of the pedestrian in relation to the final position of the car and the site of the accident
c. time-distance reconstruction of the movement of car and pedestrian, etc.

can be deemed to be accurate and reliable.

The accident files contained in the majority of cases apart from the customary police investigation results photographs of the damage to the cars and detailed medical reports on the injuries sustained by pedestrians, including autopsy reports.

Course of the Pedestrian Accident

The movements of the pedestrian through to his final position following contact with a car cannot be studied in full reality. Eye-witness accounts of the accident are due to the rapidness of events most inaccurate and unreliable.

From the dummy tests carried out by Elsholz, as well as Severy and Brink, we can, however, approximately deduce the following phases ensuing in the case of an actual pedestrian accident.

The pedestrian struck in the region of the lower extremities below the center of gravity falls with more or less high rotating velocity onto the front bonnet and into the windscreen or the side or upper windscreen frame.

If the impact does not occur at the extreme front corner of the car, resulting in the pedestrian being catapulted down one side of the vehicle, then he will remain lying for a brief moment on the front bonnet.

Due to his retardation on the car bodywork being less than the retardation of the braking vehicle, he moves forward again and drops down onto the road, where he slithers in a gyrating motion for a distance before coming to a stop.

During the final stages of motion the pedestrian passes through after leaving contact with the car, injuries of course occur, and an accurate differentiation between these injuries and those occurring during the primary phase of the accident must be left to the physicians.

Nevertheless, we may be deemed justified in associating, in particular, the severe and numerous fractures to parts of the body and simultaneous corresponding traces on the car with the contact phase between car and pedestrian.

The likewise numerous injuries to the intugement, such as haematomas, exoriations, etc., were not evaluated, as their origin was not clearly determinable from the technical aspect.

Nature and Extent of Injuries

In the accident cases investigated, the impact velocity of the car varied between 15 km/h and 90 km/h.

A notable fact, initially, was that in all these cases the pedestrian had suffered fractures to the lower extremities. Whether these fractures occurred in the region of the ankle joint, the lower leg, the knee or the thigh apparently depended on the height of the pedestrian and the height of the front bumper and edge of the front bonnet.

In almost all cases investigated, the pedestrian moved at right angles to the direction in which the car was travelling, so that one would initially have expected the fracture to have been sustained by the leg nearest to the car.

This supposition is not, however, verified by the evaluation results. In a number of cases it was found that only the leg farthest away from the car had suffered fractures, while the other leg nearest to the car showed no fractures.

The influential factor in this phenomena is presumably a question of which leg bore the main weight of the pedestrian at the moment of impact.

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The severity of the fracture, which I would categorize as a medical layman as simple fracture, complete fracture, compound fracture and comminuted fracture, apparently increases with increasing impact velocity.

In a number of cases where high impact velocity was present, fractures to both legs at the same level were observed.

Whereas, in accidents between *cars* and pedestrians fractures to the legs were basically recorded, this type of injury was, in the majority of cases, not found in accidents involving pedestrians and *vans and tramcars*, although other parts of the body, such as the skull, ribs, internal organs, etc., had sustained very severe injuries.

In accidents involving cars and pedestrians, only isolated cases were recorded of fractures to arms, ribs and pelvis and injuries to soft parts of the body and internal organs.

At an impact velocity upwards of approximately 25 km/h, however, face and skull injuries as a rule occurred. Up to an impact velocity of around 35 km/h, the resultant injuries were concussion, loss of teeth and cuts. At higher impact velocities, skull fractures and severe brain injuries resulted. The corresponding damage to the cars involved in these accidents constituted deformation of the front bonnet, splintered windscreens and depression of the roof frame or the side roof stanchions.

No significant difference in pedestrian injuries and car damage could be ascertained between accidents involving cars with a front construction as in the Volkswagen and NSU Spider and those with a front construction, such as in the case of BMW and Mercedes, for example, although one could quite well motivate a completely differing sequence of movement of the pedestrian hit.

Possibly, the examination of the material in establishing such differences was not precise enough.

The direction in which safety construction engineers must concentrate their efforts is, at all events, revealed by the results at hand showing all pedestrians hit by cars generally sustain fractures of the lower extremities and, with increasing impact velocity, severe skull and brain injuries.

SUMMARY

1. The percentage of pedestrians in the total number of traffic accident victims depends above all on the intensity of pedestrian traffic, which differs greatly from country to country.
2. It also depends to a great degree on the construction of the roads and on existing road traffic facilities, such as footpaths and pavements, crossings, signal installations, etc.
3. Pedestrian accidents are more severe than accidents involving vehicle occupants.
4. An over-proportionate percentage of pedestrian accidents, in relation to their participation in traffic, is suffered by children and old people.
5. The main danger for pedestrians ensues from crossing the road (83.2% of all pedestrian accidents!).
6. Indulgence in alcohol on the part of the pedestrian contributed to 8% of all pedestrian accidents (in the Federal Republic of Germany).
7. 7.4% of the drivers involved in serious accidents were under the influence of alcohol.
8. The hitherto available statistical data on pedestrian accidents by no means suffices to enable specific measures to be taken on an extensive scale for improving the safety of the pedestrian in traffic. This requires further detailed studies, which must approach the problem on a broad basis. German automobile insurers will carry out such studies.

Source Federal Statistics, 1970 (revised)
Subject Pedestrian Victims of Motor Vehicle Traffic Accidents
Age of Pedestrians Involved

Age		(Killed) %	Age		(Killed) %
under 6	541	8.9	25 - 34	355	5.9
6 - 9	533	8.9	35 - 44	409	6.8
10 - 14	216	3.6	45 - 54	344	5.7
15 - 17	114	1.9	55 - 64	782	12.9
18 - 20	123	2.0	65 and older	2.509	41.4
21 - 24	123	2.0	not known	7	0.1
			Total	**6,056**	**100.0**

Percentage of Pedestrians in Traffic Victims

	Traffic Victims Total	Share/Percentage of Pedestrians	
Killed	19,193 = 3.5%	6,056 = 31.6%	of those killed
Seriously Injured	164,437 = 30.0%	35,943 = 21.9%	of those seriously injured
Otherwise Injured	367,358 = 66.5%	41,506 = 11.3%	of those otherwise injured
Killed and Injured	**550,988 = 100.0%**	**83,505 = 15.2%**	**of all killed and injured**

Killed and Injured Pedestrians

Killed	6,056 = 7.3%
Seriously Injured	35,943 = 43.0%
Otherwise Injured	41,506 = 49.7%
Killed and Injured	**83,505 = 100.0%**

Accidents caused by Pedestrians These causes embrace inter alia

1	False conduct on crossings with police traffic control	48 = 0.1% (0.07)	7	False conduct on other crossings	2,697 = 3.7%
2	Bodily impediment or impairment in health	220 = 0.3% ([illegible])	8.	Under the influence of alcohol	5,753 = 8.0%
3	Not keeping to the pavement or footpath	1,169 = 1.6%	9.	Sudden emergence from behind obstructions to the field of vision	14,281 = 19.9%
4	Not keeping to the side of the road indicated	1,283 = 1.8%	10.	Other careless crossing of the road	39,945 = 55.9%
5	Playing in or alongside the road	2,589 = 3.6%	11	Other causes emanating on the part of the pedestrian	1,077 = 1.5%
6	False conduct on crossings with traffic lights	2,609 = 3.6%		**Total**	**17,680 = 100.0%**

Source Accident Statistics of the German Association of Third Party, Accident and Motor Traffic Insurers
Subject Pedestrian Victims of Motor Vehicle Traffic Accidents
Types of Accidents

1	Accident resulting from opening the door of a motor vehicle	56 = 0.2%	6.	Accidents occurring when turning right or left or making U-turns	474 = 2.5%
2	Accidents in connection with non-observance of the right of way	71 = 0.3%	7.	Accidents occurring while driving in reverse	484 = 2.6%
3	Accidents in connection with a motor vehicle being rammed by another motor vehicle from behind	80 = 0.5%	8.	Accidents arising when overtaking	557 = 2.9%
4	Accidents in connection with animals on the roadway	89 = 0.5%	9.	Accidents outside the precincts of the roadway caused by a motor vehicle leaving the road	563 = 3.0%
5	Accidents resulting from not keeping to the right-hand side of the road	202 = 1.1%	10.	Collisions between motor vehicle and pedestrian on the road	16,370 = 86.4%
				Total	**18,946 = 100.0%**

CRASHWORTHINESS SEMINAR

SECTION FOUR PART 2

OCCUPANT PROTECTION IN CAR-TO-CAR IMPACTS

RAYMOND G. FISCHER, *Sub-Project Manager*
General Motors Corporation

Much effort is currently being expended in the areas of vehicle crashes and occupant crash protection. Projects generally are in the occupant protection subsystems or deal with vehicle structures. There is, however, a need to investigate the occupant and the vehicle as an integrated system, to provide a basis for understanding the mutual effects of vehicle and occupant. Because our goal is to describe occupant performance in real crashes, a general study should not be limited to barrier impacts only.

Vehicle mix in car-to-car impacts has a measurable effect on the occupants of both vehicles. The study we are reporting today uses barrier impact performance as a basis for analytically approximating car-to-car conditions. Since the reason for safety vehicles is to provide safety for the occupants, both barrier and car-to-car impacts, using the occupant performance as a basis of comparison, have been considered.

The barrier impact presently is the only feasible and repeatable method of evaluating vehicle impacts with reasonable accuracy. If we can evaluate the trends in car-to-car impacts based on barrier impacts, then occupant protection systems can be developed using the barrier-to-barrier analysis techniques.

The objective of our study was to show both the effects on the vehicle, and the resultant effect on the occupant in car-to-car impacts for:

- Vehicles of equal barrier performance but different masses, (this means that the time-velocity history and crush are equal).
- Vehicles of different barrier deceleration characteristics but equal crush and mass.

Looking first at mass differences, vehicle effects were evaluated as a result of the change in occupant performance due to vehicle changes.

Occupant restraint systems have a definite effect on the occupant performance in the vehicle. I would like to show you a simplified theoretical approach to this problem.

Figure 1 shows a graph of occupant velocity vs time for a constant occupant deceleration. The distance travelled is then the area under the curve. His unrestrained flight is shown as a constant velocity. A belted occupant is shown with 2 inches of free flight (A). This would represent a restraint system on an impact sled, which is stopped instantaneously (i.e., rigid car with infinite deceleration).

If we assume a 42 g occupant deceleration, with an initial velocity of 30 mph, and 2 inches initial clearance before his restraint system is activated, we can see from this graph that he travels the initial distance (A) in about 4 milliseconds. The restraint is then applied, (maintaining 42 g) until he comes to rest (B). This takes an additional 33 msec and requires 8.4 inches more travel, totaling about 10.5 inches. This is probably the most efficient system that can be devised for minimum space, and represents a lap and shoulder belt, with no vehicle around the occupant. The 42 g deceleration was chosen to represent a 60 g system with 70% efficiency, which is the best we could expect to obtain with real hardware.

The occupant velocity curve, shown as AB^1, is the result of applying a 1,500 g/sec maximum change of acceleration limitation to the restraint system. This increases the crush required by about 70% (Figure 2).

Referring to the *14th Stapp Car Crash Conference*, provides values of tolerable jerk calculated by Col.

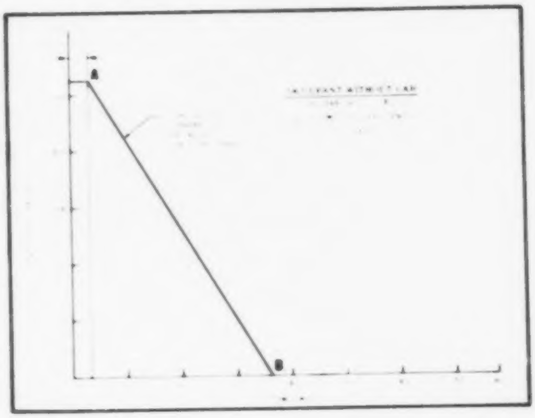

Figure 1

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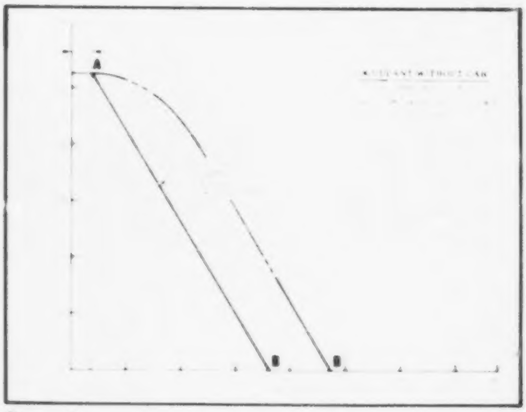

Figure 2

Stapp in the range of 20,000 to 50,000 g's sec. This would cause negligible additional crush, and curve AB can be used for this discussion, ignoring the small onset curve, which would normally be provided by the occupant's compliance in a real life impact.

If we look at the same situation substituting a theoretical air cushion for the close coupled restraint system (Figure 3), we see that the initial distance travelled (C) is dependent on the time required for developing its effective force (includes firing, inflation time, and initial load onset). Although the penetration required is the same (C to D), the initial clearance is increased, making the total travel about 32 inches.

Since the restraint load (42 g's) is applied to both occupants for the same time, the severity index is constant for both. Since the efficiency is included in the restraint g's, the severity index we could expect is based on a 60 g restraint, and is calculated at 635.

What are the effects of adding a vehicle to the occupant subsystem?

Figure 4 shows us what happens. The occupant in this case is placed in a rather stiff, 27 g square wave vehicle (representing a square wave vehicle deceleration in a 30 mph barrier impact). The close coupled restraint is applied when relative occupant-to-vehicle displacement reaches 2 inches (A), and the air cushion is again assumed to be effective at 45 msec (C).

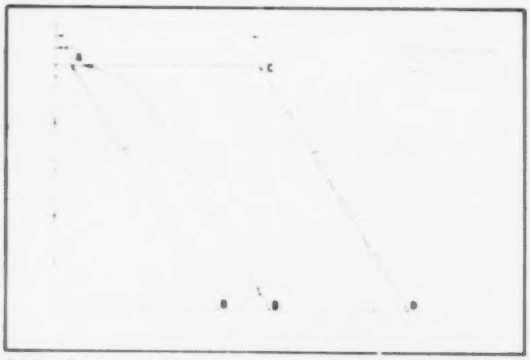

Figure 3

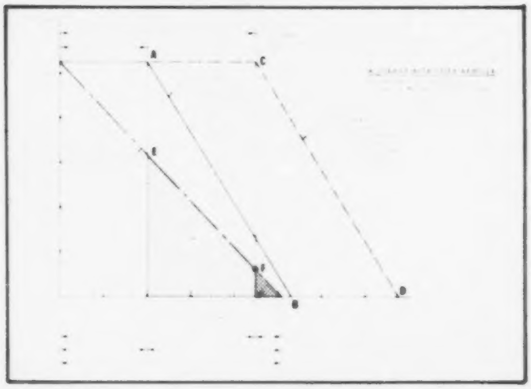

Figure 4

Note that in this case it takes 20 msec to take up the 2 inches slack in the belt system, while the air cushion effectiveness is based on time only.

Since the vehicle reaches 0 velocity before the occupant in both cases, the SI remains unchanged at 635.

With the close coupled restraint, the occupant engages his restraint after the vehicle has crushed about 8 inches (E), providing some ridedown (shown by the shaded area) while the vehicle crushes the remainder of its 13.5 inches of crush. This results in a reduction of interior occupant displacement required from the previous 8.4 inches to 3.5 inches resulting in a total travel of 19 inches, but requiring only 5.5 inches inside the car instead of the 10.4 inches with no vehicle action.

In the case of the air cushion, the free flight distance (C) is reduced to 10.5 inches with respect to the vehicle interior, but most of the penetration (8.2 inches) is still required to stop the occupant. He contacts the vehicle at (F), and very little ridedown is gained, as shown by the small area, resulting in a total travel of 32.2 inches with respect to a space fixed reference, identical to the case where no vehicle was used.

In this next case, it is interesting to observe what happens when the vehicle is replaced by a softer vehicle (Figure 5). The new vehicle is a 17 g square wave, with the same restraint systems as before. The lap and shoulder belt restraint (A) still requires 2 inches of occupant displacement, but the belt force is applied for a shorter time to bring the occupant-to-

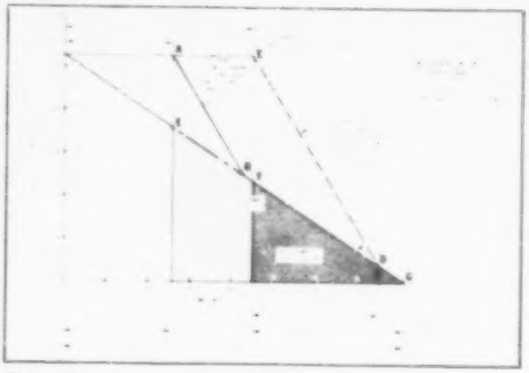

Figure 5

vehicle velocity. This reduces the SI to 330. The SI is calculated based on the restraint applied time until the occupant reaches vehicle velocity, plus the SI during vehicle ridedown.

The occupant interior displacement is reduced to 1.5 inches, giving us an occupant-to-vehicle travel of 3.5 inches after the vehicle has crushed 11.7 of its 21½ inches of total crush. The ridedown area is significantly increased, but the total occupant travel is now increased to about 25 inches compared to 19 inches in the stiff vehicle. This is probably not unreasonable considering the improvement in severity index.

The air cushion (C) also showed some improvement, as some ridedown benefit is gained as shown in the area under FG. The initial occupant-to-vehicle travel is decreased to 6½ inches, the penetration to 4½ inches. The severity index is reduced to 512. The total travel of the system, however, has changed little.

This analysis leads us to the conclusion that, assuming a tight belt restraint system is used, proper control of the interior space (free flight distance and "crush distance") and of the vehicle structure can provide a reasonable margin for designing a *safe* level of occupant protection for all sizes of vehicles currently in use. Obviously, this analysis is based on theoretical considerations associated with ESV's, and practicality has not been addressed.

At this point, I would like to discuss briefly the results of studies to determine the effects of mass differences and barrier g-time characteristics of vehicles on the occupants.

In a diffuse study of this type, it was expedient to develop a simplified computer model capable of handling barrier impacts, car-to-car impacts, and the resultant occupant performance with minimal specific input required. While this provides an approximate solution, limited by the simplifying assumptions, it can be used to study trends and for preliminary parameter studies for new vehicle proposals.

The approximate car-to-car impact analysis model (Figure 6) can perform the following functions:

- Calculate the vehicle velocity, displacement, crush, potential energy and static and dynamic barrier force functions for a complete barrier impact for two vehicles simultaneously.
- Calculate the mutual and individual car-to-car impact characteristics (acceleration, time, velocity, displacement, force and crush) for the two subject vehicles for any relative velocity, limited by the static force and crush generated in the barrier calculation.
- Calculate the relative impact velocity and time of impact of a simple occupant-to-vehicle, occupant crush, occupant velocity, and occupant severity index for all conditions.

The input is the desired or measured barrier deceleration time history of the vehicle, and its mass. The model is capable of handling multiple occupants, differing in location with respect to their restraint system. Occupant deceleration is input as a square wave deceleration desired.

The Effect Of Varying Vehicle Mass

A study to determine the mass effect of vehicles in head-on car-to-car impacts was undertaken for a relative impact velocity of 60 mph. A 31.1 g peak sine wave barrier pulse was selected for the vehicle input.

It was assumed that:

- The 40 mph barrier performance of all vehicles was identical.
- Occupant deceleration is 60 g's for all cases.
- Vehicle mass (and its associated barrier force) is the independent variable.
- It is assumed by the simulation that the vehicles will match in a car-to-car impact.

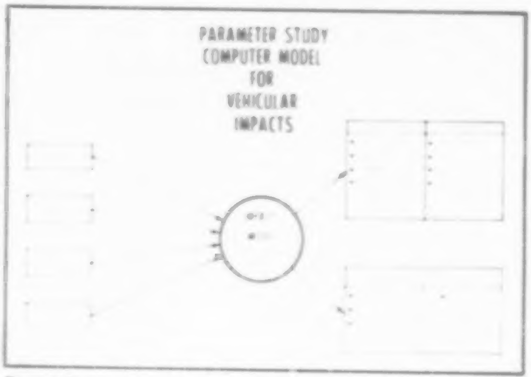

Figure 6

In addition, three occupant-to-restraint initial spacings were assumed — 2 inches, 6 inches, and 12 inches. This provided chest severity index for assumed close coupled (i.e. belts), active (i.e. air cushion), and passive restraint variation concurrently with the mass effect study.

Impacting two identical vehicles (mass ratio = 1) would be the barrier equivalent of 30 mph. Figure 7 shows the computer output for the vehicle 40 mph barrier calculations. The Y axis is used to indicate acceleration (g's), velocity (mph) and displacement (in.) plotted against time in milliseconds (Figure 8). This provides a capsule view of the vehicle. The acceleration is shown by the solid line, the velocity by the dashed line, and the displacement by the curve indicated with squares (Figure 9). The sine wave vehicle acceleration was used to study the effect of vehicle mass on the occupants.

The vehicle weights were varied from 2,000 to 5,000 pounds, and results were obtained in terms of mass ratios varying from 0.5 to 2.0. The net vehicle velocity change, vehicle rebound velocity, vehicle crush, occupant relative impact velocity and occupant chest severity index are then plotted against the mass ratio of the vehicle being evaluated to the other impacting vehicle. The trends, based on occupant performance, can then be established.

The effect of the mass differential is shown in Figure 10. The severity index is plotted on the Y axis, and the mass ratio of the subject vehicle to the impacted vehicle on the X axis. Occupant severity for Car 1 is plotted vs mass ratio for three initial spacings, the 2 inch spacing (indicated by A), 6 inch spacing (indicated by B) and 12 inch (indicated by C) for

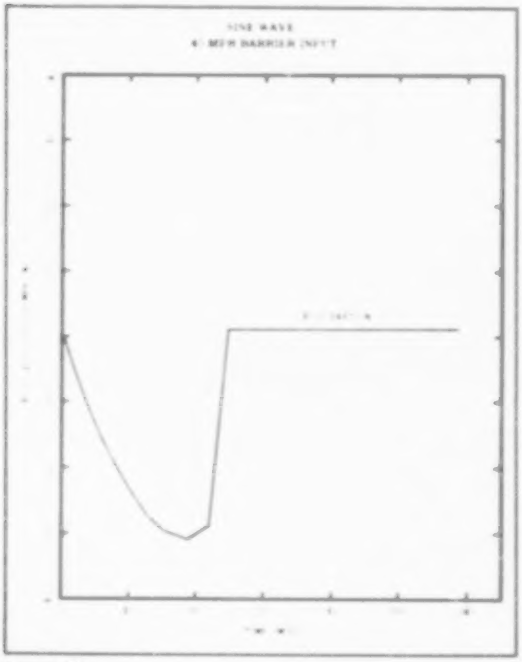

Figure 7

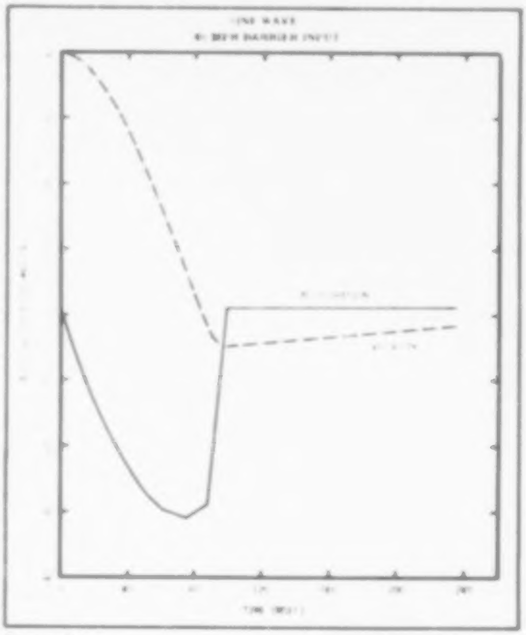

Figure 8

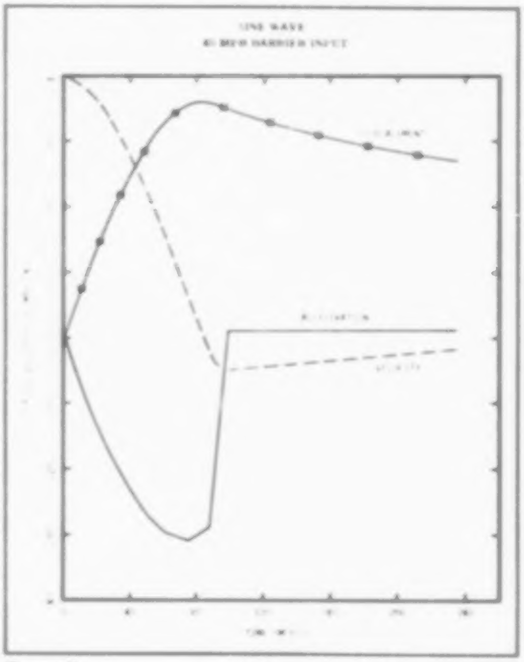

Figure 9

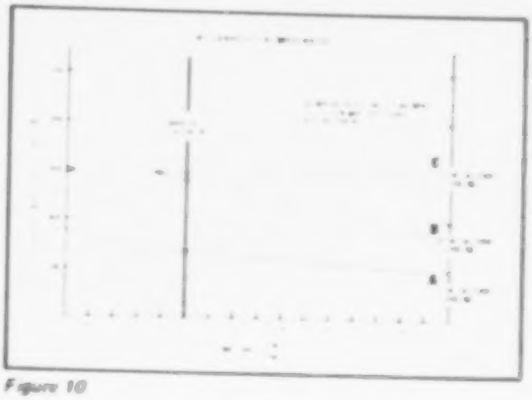

Figure 10

occupants in the vehicle being studied. These results show that a series of computations for various mass ratios can establish a trend.

Using the barrier equivalent impact speed (mass ratio = 1) as a base, the following conclusions can be generalized:

- The restraint system influence is more important than mass ratio, as can be observed by looking at curve A. The increase in SI from a mass ratio of 2.0 to 0.5 is about 150. If we compare A and C for the 2.0 mass ratio, we see a change of almost 400 in SI. This trend can be observed for all cases.
- An increase in the mass of a vehicle over an impacting vehicle results in improvement in occupant performance. As occupant spacing increases, the SI reaches a limiting value. This limit, as shown in C, occurs as the vehicle ridedown available to the occupant is reduced.

The occupant impact velocity also varies with mass and spacing. This chart indicates the relative impact velocity of the occupant-to-restraint system, shown on the vertical axis. This is plotted against the mass ratio (on the horizontal axis) for the three occupant spacings of 2 inches, 6 inches, and 12 inches, labeled A, B, and C respectively. Comparing the change of velocity for each spacing for the range of mass ratios, to the change caused by an increase in spacing, it becomes evident that spacing is again more important than mass ratio for restrained occupants.

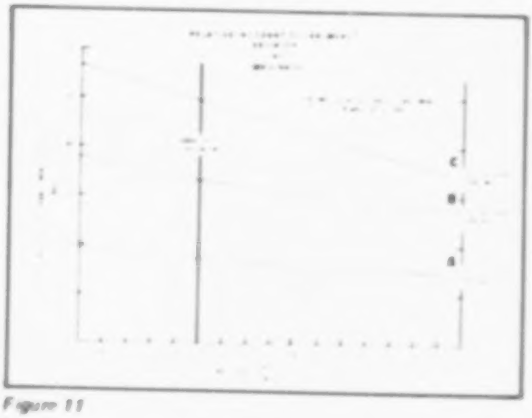

Figure 11

For vehicles with the same barrier characteristics and different masses:

The heavier vehicle will crush less than the lighter due to additional structural force capability. Figure 12 illustrates the effect of the crush ratio, which is the ratio of the crush of the two respective vehicles, plotted against the mass ratio (horizontal axis) of Vehicle 1. The crush ratio indicates directly the percentage of the total mutual crush of the two vehicles which will be sustained by the subject vehicle impacting another vehicle.

The total velocity change of a heavier vehicle striking a lighter vehicle is less for the heavier vehicle. Figure 13 illustrates the effect. Again, mass ratio is plotted on the horizontal axis and the net velocity change is plotted on the right hand vertical axis. With

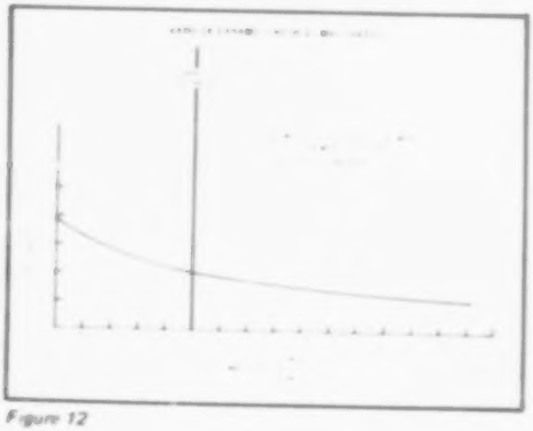

Figure 12

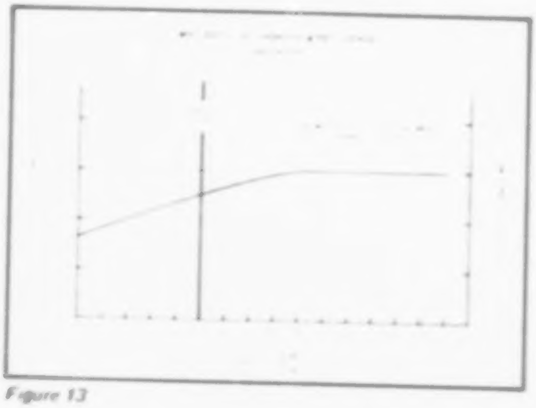

Figure 13

each vehicle travelling in opposing directions at 30 mph, it can be seen that the heavier vehicle, i.e. 2.0 mass ratio, has a change of 30 mph, while a lighter vehicle, i.e. 0.5 mass ratio, has a change of 44 mph. This is an effective rebound velocity, shown on the left side, of 14 mph for the lighter vehicle (labeled **REBOUND VELOCITY**).

We can conclude in this example that the lighter vehicle (.5 mass ratio) sustains all of the rebound from the stored energy in the system. The barrier impact rebound velocity of the vehicle would be as shown for a mass ratio of 1.0, about 6 mph, which depends on barrier rebound velocity.

Vehicle Compatibility Study

A study of the compatibility of vehicles with different g-t characteristics, based on occupant severity, was a logical step from the mass difference trend study. Assuming:

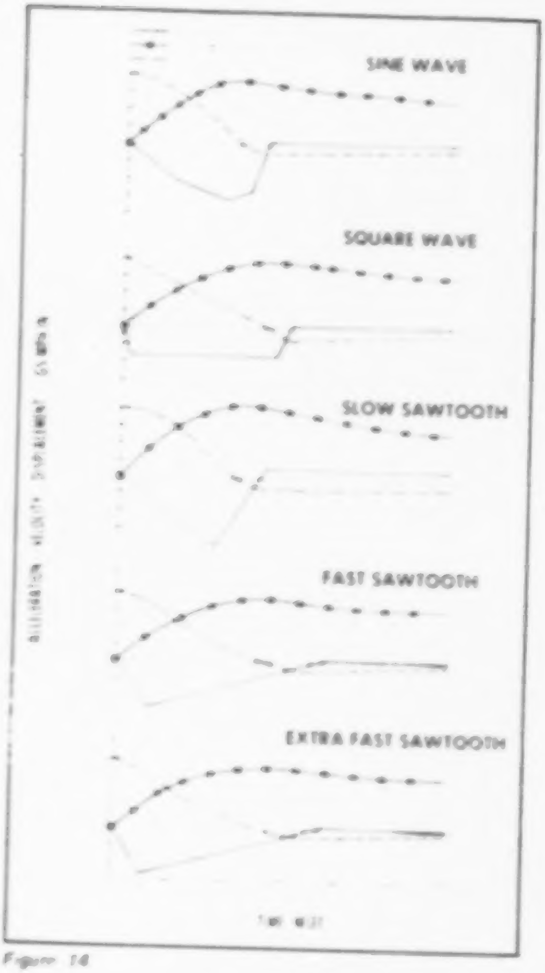

Figure 14

- The mass effect still holds, thus the variation in wave shape of the vehicles' barrier acceleration time history can be superimposed.
- The vehicle to barrier crush and rebound velocity are essentially the same for all vehicles at 40 mph with the variable being acceleration-time only.
- Vehicle masses are identical.
- The occupant initial clearance assumed was 2 inches, 6 inches, and 12 inches as previously discussed.

Acceleration-time input curves were constructed so that essentially equal energy was dissipated for all vehicles to meet the requirements above (Figure 14). In order to study a range of conditions, the curves selected were: sine wave, square wave, a slow rise time sawtooth and two fast rise time sawtooth curves.

After the evaluation of each vehicle against the others at a relative impact speed of 60 mph, and against itself (30 mph barrier equivalent), a means of comparing the vehicles was devised (Figure 15).

The technique used for the comparison ranks the vehicles tested in an inverse order of average severity index for occupant spacings tested (highest number is the lowest SI). These rankings are arranged into matrix form, and the rank of any vehicles impacting another vehicle, including itself (barrier equivalent - 30 mph) can be determined. This ranking is little affected by occupant spacing.

It can be concluded from this conceptual study that:

- Vehicle-to-barrier acceleration-time wave shape has an effect on SI in car-to-car crashes.

INPUT G-T	SQUARE WAVE	SINE WAVE	SLOW SAWTOOTH	FAST SAWTOOTH	EXTRA FAST SAWTOOTH
SQUARE WAVE	15	14	9	11	12
SINE WAVE		13	7	8	5
SLOW SAWTOOTH			10	1	4
FAST SAWTOOTH				3	6
EXTRA FAST SAWTOOTH					2

Figure 15

- The square wave acceleration-time wave shape tends to be the most compatible (with the others studied).

If we assume that passenger compartment integrity can be maintained, we can draw the following main conclusions from our manipulations of Newton's equations.

- The design of the restraint system is the one factor that can have the greatest influence on the occupant.
- The occupant restraint system must be matched to the vehicle in which it is to be used.
- And finally, on the basis that reducing the barrier capability of heavy cars does not solve the problem, the occupant restraint system of the light cars must be the equalizing factor to provide for compatibility in a traffic mix of light and heavy cars. While restraint system design in light cars will be difficult and perhaps limited to belt systems such arrangements offer a reasonable and possible solution.

CRASHWORTHINESS SEMINAR

SECTION FOUR

PART 2

CAR-TO-CAR COMPATIBILITY

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INTRODUCTION

As reported in the U. S. presentation here at Kyoto, the initial American test results on car-to-car compatibility are beginning to become available. European compatibility tests have been reported at earlier conferences (1, 2, 3, 4)*. The compatibility problem has caused considerable controversy in the United States and to a lesser extent throughout the international ESV program. Therefore, it seems appropriate to pause at this point and briefly review the events of the last five years from the standpoint of the U. S. ESV program. Our purpose in so doing is to stimulate further discussions leading to a better mutual understanding, even though our base may change as you present your comments and criticisms. In inviting a continuation of our give-and-take interchange, we hope to establish even better uniformity in our communications.

Two basic approaches to front end energy management are being considered in the United States for lessening car-to-car impact severity. These are (a) force regulating systems such as hydraulics, either with fixed force backup structure (hybrids), or without, and (b) force ramp designs. Either system is capable of providing energy absorption, to varying degrees, in the striking car at force levels below the maximum design barrier impact force. Both systems suffer from disadvantages which will be discussed later. At this point, we merely wish to establish (a) that two techniques do exist, (b) that neither technique appears to be a panacea, (c) that further research is needed on both techniques and (d) that our end goal should be a situation where the automotive manufacturer chooses the system, or combination of systems, that best fulfills the needs of his particular vehicle.

HISTORY

Historically, no review of car-to-car compatibility would be complete without reference to the work done at the University of Minnesota in the 1950's which was initiated under the auspices of Colonel John Paul Stapp of the U. S. Air Force (5). The emphasis in this work was on reduction in crash forces in single vehicle impacts, but these findings led directly to consideration of the two car problem on the U. S. ESV program in 1968 and 1969. As a result of the ESV efforts in the late 1960's, two concepts for front end energy management were documented by the ESV Phase I evaluation contractor in February 1970 (6). These were the two concepts mentioned earlier, hydraulic and force ramp systems.

The foregoing brief narrative brings us to the issuance of the Family Sedan Specification on February 17, 1970. Because of the unpopularity of the subject of car-to-car compatibility in the U. S. at that time, it was decided not to stress it in the text of the specification. Instead, the limits on passenger compartment acceleration were specified. Obviously, these limits were on a "best estimate" basis with the goal of achieving car-to-car compatibility with recognition that the so-called barrier equivalent velocity is low for a significant number of car-to-car crashes.

The resultant specification was discussed in Paris in 1971 where our goals of achieving car-to-car compatibility were again documented (7). It was then that we became aware that the French were independently studying the problem and had already conducted some tests (1). As an aside, the French introduced the term "aggressivity" at that meeting and the term, with some variations, has been in common use since. An in-depth theoretical study of hybrid systems was conducted by the Germans in 1971 and 1972 (8, 9). All studies have indicated that some measure of compatibility can be achieved and attention is now turning to determining how much compatibility can be obtained at what cost (3, 4, 10). Because of the extreme complexity involved in the broad range of weights, configurations and types of structures, it is not anticipated that a simple early solution will be forthcoming.

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* Numbers in parenthesis refer to references at the end of the paper.

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SIMPLIFIED BASIC CONSIDERATIONS

While the treatment of a complex subject in simplified terms is always dangerous, in this case it is believed to be justified. A common basic starting point is needed for clear communication even though this simple base requires modification as second and third order effects are introduced in more rigorous considerations. Therefore, the following is presented for your comments and criticism which will nudge us toward a common foundation in the future even though it may be different from the material presented here. Some of the ruling laws in a two car collision are as follows:

1. From Newton's third law, the interface forces are equal.
2. From Newton's second and third laws, the accelerations in the two vehicles will be in the inverse proportion to the masses.
3. To the first approximation, the velocity changes in the two vehicles are also in inverse proportion to the masses. This follows from the fundamental relationship that the change in velocity is equal to the integral of acceleration over time. In a car-to-car crash, both cars are subjected to the same acceleration period if outside forces are neglected.

Statement 1 concerning equal forces is generally recognized, but it is sometimes forgotten that the more compliant structure sets this force level. In practice, this usually means that the force level is determined by the smaller of the two colliding vehicles although this is not always true. When it is true, it is incorrect to state that the big vehicle exerted too much force on the smaller as long as the smaller is deforming at essentially design force. A better statement is that the large vehicle exerted the design crush force of the smaller vehicle upon it without deforming; therefore, all of the small vehicle crush distance was used before the required amount of energy was absorbed. Note that in theory, if the large vehicle had also been deformed at this force level, part of the energy would have been absorbed in this process. If this had occurred, the energy absorbed in the big car could then have been subtracted from the total energy absorption requirement, thereby reducing the stroke required in the small car.

Statement 2 concerning acceleration ratios is often cited as a reason that the small car must be subjected to extremely high accelerations. This is incorrect. While the ratio of the accelerations is fixed, the absolute values of the accelerations can be controlled within limits if force and time can be controlled. For example, assume a mass ratio of 3 to 1 between two vehicles. In theory, the accelerations of the large vehicle to the small can be 10 g/30 g, 15 g/45 g, 20 g/60 g, etc., depending on the designs of the two vehicles.

Statement 3 concerning the ratio of velocity changes is theoretically inflexible if it is assumed that the coefficient of restitution is constant and that no outside forces are acting on the two body system. The magnitude of the velocity change each vehicle must experience is theoretically fixed from the laws of conservation of energy and momentum. However, if this velocity change can be made to occur over a sufficient time period (at a low enough acceleration, as discussed in considering Statement 2), the consequences of the large velocity change for the small car can be ameliorated.

We may summarize at this point. If we assume a constant force throughout the deformation in both cars in a two car crash at a given impact velocity:

a. The force level is set by the "weakest" car.
b. The energy absorption requirement is determined by the difference in kinetic energy before and after the crash. This difference in energy must be dissipated as work.
c. From the definition of work, i.e., the integral of force over distance, a specific deformation distance is required since the force level is fixed.
d. This required deformation can all occur in one of the cars or it can be shared in the two cars if they have the proper force deformation characteristics.
e. Considering big car/little car compatibility, all energy absorption that occurs in the big car (at the proper force) can be directly substracted from the small car absorption requirement.

This discussion has dealt only with frontal impacts to this point. On the road, partial frontal engagement, angular crashes from the front, side and rear, right angle side impacts and straight-in crashes from the rear predominate. All of these are included in the subject of car-to-car compatibility. An exhaustive treatment of all these crash modes, even on a basic level, would be beyond the scope of this partial review. Therefore, only side impacts will be briefly discussed because of their significance in causing serious injuries and fatalities. However, it is clear that the other crash modes must be addressed before the total subject can be considered to have been dealt with even on a preliminary and tentative basis.

In the case of side impact, there is a severe limitation on crush distance in the struck automobile. This limitation is off-set somewhat by the force-limiting nature of the interaction due to the movement of the struck vehicle away from the striking car.

Two level reactions, spreading the load over a larger area and providing sill-to-bumper load paths have all been proposed and demonstrated at what appear to be reasonable impact velocities. However, much remains to be done before complete system performance will have been demonstrated. To mention only one facet of the total system, pedestrian protection must be integrated into vehicle front end performance along with side impact capability.

It is obvious from mechanics that much of this reasoning can also be developed from the standpoint of impulse and momentum since both energy and momentum must be conserved. It is sufficient here to say that time, in conjunction with force, appears in the expression for impulse in the same manner that it appears in conjunction with acceleration in the expression for velocity.

A further problem which has not been addressed to any extent in this paper is restraint compatibility with structural crush. This may be an extremely serious problem, particularly in a small car which is first stopped and then propelled backwards in a big car small car collision when the two cars impact at equal velocities. Obviously, if the impact time is lengthened by providing additional aggressiveness control stroke in the big car, the occupants of the small car will fare better. Clearly, this problem needs further study on both an analytical and empirical systems engineering basis.

SPECIFIC CONCEPTS

Each of the concepts discussed in this section is amenable to modifications and could assume many forms with altered response characteristics. Obviously all of the possible combinations and permutations cannot be considered within the scope of this paper. However, it is believed that a brief look at some of the basic concepts may be useful in leading us toward mutual understanding of the several engineering approaches that are being followed.

Fixed Force

In this concept, the force rises rapidly to the design level and thereafter remains approximately constant throughout the deformation. Assuming zero rise time (which is impossible on a practical basis) the deformation varies as the square of the barrier impact velocity, while the impact time varies with the first power of the barrier impact velocity. In its pure form, it appears that extreme difficulties are posed in matching fixed deformation forces in two (or all) cars so that deformation, and therefore energy absorption, will be shared in car-to-car impacts. Also, restraint compatibility may pose problems due to the short deformation stroke. However, fixed force structure can be used efficiently for force limiting backup structure behind ramps or variable force systems. Such force limiting may be desirable for all size cars in high speed barrier impacts and for smaller cars when struck by less compliant vehicles. The producibility of the structure with approximate fixed force response over reasonable stroke lengths is generally considered good.

Ramped Fixed Force

In this concept, the maximum force level is reached gradually over an appreciable distance. If a straight line ramp is assumed, it can easily be shown that the maximum efficiency of the ramped portion cannot exceed 50 percent; i.e., the ramp deformation distance is theoretically only 50 percent utilized. In addition, the deformation distance is determined by the force. Therefore, the energy absorption of the ramp is decreased not only by the 50 percent efficiency but also by the shorter stroke in low force impacts. However, the ramped stroke does absorb some energy at all force levels, though the efficiency is low, and therefore the concept can be used to achieve some measure of car-to-car compatibility. It also provides sufficient stroke (or delay) for restraint compatibility over a reasonable range of impact velocities. Producibility is still an open question but it should be noted that the structural response of conventional vehicles has been reported to be approximately a force ramp (11). A tradeoff with body damage in low and intermediate speed impacts may also be required. In addition, it may be necessary to relocate the engine in front engine cars or provide for engine deflection in high speed impacts.

Hydraulic Systems

Hydraulic systems are extremely versatile as far as force response characteristics are concerned. However, in the interests of brevity and simplicity only one type of system will be discussed in any detail here. This is the fixed stroke system. This system has been criticized because of its full stroke characteristic at all force levels which has been regarded as causing excessive body damage at low speed. It has been shown that this characteristic can be eliminated and the reader is referred to the Fairchild and AMF designs for details noting, however, that the AMF design is not a fixed stroke system (12, 13, 14). Since the stroke or deformation is fixed, the force level in a pure fixed stroke system varies with the square of the barrier impact velocity. With regard to impact duration, the time varies inversely with the first power of the barrier impact velocity. Therefore, the barrier impact duration is long at low and intermediate speeds where many car-to-car impacts occur. The implications of this have not been empirically investigated for car-to-car collisions. Unfortunately, from the standpoint of communication, the device was initially designated as being velocity sensitive. While this is not incorrect, the reaction force is also regulated by the mass and stiffness of the struck object. Therefore, when a small vehicle is struck by a hydraulically equipped large vehicle, the large vehicle will stroke at the interface force set by the small vehicle. From the previous discussion, and still using the same assumptions, it can be seen that this energy can be subtracted from the total energy absorption requirement for the two vehicles to get the absorption required in the small car. Note that this capability of stroking at low force, when needed, does not interfere with the high speed barrier capability of the system since the force is self regulating.

Turning now to efficiency, the stroke efficiency (not to be confused with the efficiency during the stroke) of hydraulic cylinders is slightly less than 50 percent. In other words, the cylinder does not normally crush. This causes varying amounts of design difficulty depending on whether the cylinder can be recessed and/or moved rearward. The hydraulic stroke itself is very efficient. Therefore, if the cylinder can be recessed, the efficiency is normally in the range of 80 to 90 percent. This high efficiency may lead to restraint compatibility problems at higher impact velocities. It is obvious that either lengthening the stroke at constant force or "spoiling" the efficiency by some other means are tentative solutions to the problem. In either case, the total stroke will increase.

Hydraulic energy absorbers built to date have been heavy. However, there has been no serious attempt to reduce weight. The AMF hydraulic systems have been shown to be capable of multiple high speed barrier impacts. The same is believed to be true of the Fairchild systems. This is a margin of safety that is not needed by the consumer. Until systems are built that fail in a safe manner at high impact speeds and serious weight reduction programs are conducted, the final weight of these systems will remain conjecture.

From a producibility standpoint, hydraulic systems are more complex than simple crush structure. However, they compare favorably with hydraulic brake systems which were adopted to get better self-equalizing braking performance. In comparison to automatic transmissions, which are only a convenience item, they are much, much simpler. However, it may be that these systems will find acceptance, if at all, in hybrid system.

Hybrid Systems

As used here, a hybrid system is one in which the initial portion of the stroke is self-regulating; i.e., hydraulic in practical terms today, and the later portion is fixed force. In this concept the self-regulating portion of the stroke is the aggressiveness control while the later portion is force limiting and provides additional crush for high speed barrier impacts and collisions with less compliant vehicles. In this concept, the inverse relationship of time with impact speed of the self-regulating portion is stopped and "turned around" by the fixed force portion of the structure when it begins to deform. If this concept is used to achieve compatibility the lightest car needs no self-regulating stroke for compatibility and the self-regulating stroke requirement increases with the mass of the vehicle. The German studies which were referred to earlier (8, 9) are on such a system designed for 75 mph closure velocity. It is clear that other closure velocities, either fixed or variable with the mass of the car, can be used for the design criterion.

The efficiency of this concept is theoretically high. Designs that "bleed off" the self-regulating stroke at forces below the fixed force level are less efficient and sacrifice some of the "aggressiveness control" range. It is possible to encounter restraint compatibility problems with this system also and potential solutions are the same as those for the fixed force or hydraulic systems. With regard to producibility, it is clear that the earlier statements about fixed force and hydraulic systems both apply.

CONCLUSIONS

1. Aggressiveness is a significant problem in mixed traffic.
2. Concepts have been developed, and in some cases demonstrated, for partially alleviating the problem.
3. It is not anticipated that there is a simple, single solution.
4. In most cases, the larger car (assuming it would be least compliant) must make provisions for accommodating the smaller car.
5. At present, much remains to be done in defining the tradeoffs required by the candidate systems.
6. One system, the force ramp, uses excessive distance but is of more conventional construction.

7. Another system, hydraulics, permits short close-coupled designs but adds complexity.
8. Many real world car-to-car crash conditions have received only superficial attention to date.
9. Restraint compatibility with structural response requires much more study for all cars in high speed impacts and small cars pose special problems in car-to-car impacts as well as in barrier crashes.
10. The eventual goal should be to provide requirements that allow the manufacturer to choose either system, or other as yet unidentified systems, that is best-suited for his overall design. However, car-to-car compatibility must be achieved.

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SECTION FIVE

FUTURE SAFETY STANDARDS AND THE ESV PROGRAM

PART 1 THE UNITED STATES PRESENTATION

DR. GENE G. MANNELLA
NHTSA R&D Program — An Overview

PART 2 PRESENTATIONS BY THE COMMITTEE OF COMMON MARKET CONSTRUCTORS (CCMC)

M. OUIN
Introduction

DR. G. M. MACKAY
Field Studies of Traffic Accidents in Europe

DR. CLAUDE TARRIERE
Efficiency of the 3 Point Belt in Real Accidents

DR. GUNTHER BRENKEN
ESV — Development and Legislation

PART 3 THE JAPANESE PRESENTATION

HISASHI KAGEYAMA
Future Safety Standards for Japan

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SECTION FIVE

PART 1

NHTSA R&D PROGRAM – AN OVERVIEW

DR. GENE G. MANNELLA, *Associate Administrator for Research and Development*
Research Institute
National Highway Traffic Safety Administration

During the course of this conference, as well as during the preceding ESV conferences, many of the technical presentations have quite correctly stressed the urgent need for fundamental research in a variety of areas. Important items such as biomechanics, restraint systems development, and accident analyses have received particular mention.

Within the NHTSA research program, these and many other important problem areas are studied by offices and divisions outside of the ESV office. In order to provide a more balanced and well-rounded view of the total safety research and development being carried out by NHTSA, it seems appropriate at this time to highlight, briefly, some of the important efforts now underway in the Research Institute which complement the ESV program.

In addition to the Office of ESV Programs, the Research Institute is comprised of four other offices:

- Office of Operating Systems Research
- Office of Vehicle Structures Research
- Office of Driver Performance Research
- Office of Accident Investigation and Data Analysis

As their names imply, each office is focused on a major critical area of safety research. Considering each of these offices in turn, I would like to briefly highlight some of the more important aspects of their work.

First, in the Office of Operating Systems, major R&D programs have been underway for several years in vehicle handling, braking, and vehicles-in-use. These programs include fundamental, single factor analyses (such as research in brake fluid composition and performance) to complex, multiple systems studies (such as the mathematical modeling of total vehicle dynamics). In our research on vehicle performance properties, we have studied not only *new* vehicle designs and operating characteristics, but also the important problem of vehicle systems degradation and wear. The safety consequences of the gradual and inevitable change in vehicle systems' performance under varying usage environments are important, not only to the establishment of effective programs for keeping unsafe vehicles off the road through inspection and regulation, but also in prescribing important initial design characteristics for new vehicles, including ESV's.

In the Office of Driver Performance, we are conducting research in the companion areas of lighting and visibility. Particular attention has been devoted to improving both rear and forward lighting systems. In the area of rear lighting, for example, our studies have concluded that the initial tendency to develop more complex coding and signalling patterns (with the associated penalties in increased system costs and engineering complexities) is, for the most part, unrewarding in terms of measurable safety performance improvements. Simple functional separation of some signals appears to be a promising approach, while the safety value of systems using several different colors remains unproven.

The areas of direct (forward) and indirect (rearward) visibility have also received considerable attention. We are learning more in these areas every day, and we anticipate that the results of current efforts will permit significant refinement and improvement of the visibility requirements portion of the 3,000 pound Research Safety Vehicle before the final specifications are established.

The Office of Vehicle Structures is responsible for our crash survivability research program covering research in biomechanics, occupant packaging, and structures. The biomechanics research program broadly addresses the areas of vehicle occupants, motorcyclists, and pedestrians. The common problems in these areas are the determination of human tolerance to impact, and application of this knowledge to subsystem design and human simulation. The biomechanics effort supports all elements of crash survivability and is considered the key or pivotal element in our supporting research program. Within the limits of our resources, we have been investigating injury modes and tolerance to the head, chest, pelvis/femur, and abdomen in that order of priority established by accident statistics. As these data become available, it is fed to our human crash injury computer model which we expect will eventu-

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to provide the design criteria for occupant protection and human simulation systems.

We are currently contemplating a long-range attack on the human tolerance program to accumulate the much-needed body of knowledge associated with human response to contact and dynamic forces. The subsequent and/or parallel development of a complete family of standard anthropomorphic dummies for use in all transportation research and compliance testing is a distinct future possibility.

Our work in vehicle structures has been primarily aimed at improving the structural crashworthiness of production vehicles as a part of a baseline investigation of crash energy management at reasonable weight increases. This work has included barrier and car-to-car crash investigations over a wide variety of crash modes and speeds, and has provided much basic data to many of the ESV participants. In the immediate future, this work will be expanded to include considerations of aggressivity for vehicles of various mass.

As you know, our recent work in occupant restraint systems has concentrated heavily on inflatable restraint systems over a range of vehicle speeds, vehicle sizes, and passenger positions. We are also conducting an evaluation of all available passive belts systems, and on the basis of the results of this evaluation, expect to develop some of the more promising systems to a higher level of performance in the hope that there will be a variety of options available to the public which meet current and projected standards. We will also devote attention to seat design and other elements of interior protection which must be considered in conjunction with restraint systems design in obtaining optimum occupant protection.

Research in driver and pedestrian factors is the responsibility of another office-level group within the Institute. Driver behavior under both normal and adverse driving conditions is studied in-depth. Human performance capabilities and limitations are defined to permit better design of controls, displays and warning systems. The term "driver control," which embraces all aspects of enhancing or restricting driver actions, is emerging as a target NHTSA action area requiring considerable research attention. Education and licensing requirements, for example, must be based on a sound technical information base, not just tradition or intuition.

Primary attention has been given by this office to alcohol counter-measures research, including the development and testing of techniques and devices for the rapid, on-the-spot, accurate determination of driver blood alcohol levels.

Pedestrian safety was the subject of a recently completed special task force study effort. As a result, renewed vigor is being injected into coordinated research and action programs to attack this increasingly important problem. In addition to these efforts, the recently completed Pedestrian Safety Project under CCMS has produced important recommendations to be integrated into our ESV development. Success here will not come easy, and although we have devoted some attention to this in our 3,000 pound RSV specifications, much remains to be done before a final optimum requirement can be formulated.

In contrast to the "research laboratory" character of the work described above, the results of the efforts of the Office of Accident Investigation and Data Analysis are derived from the real-world operating environment of drivers, passengers, pedestrians, vehicles and the highways. The urgent need for a better understanding of what happens in *real* crashes, as well as where, why, and how often it happens is evident to all of us. We heartily endorse the remarks of several of our speakers this week in calling for better accident data and increased accident data analysis.

Major programs in both of these areas have been underway in the U.S. for several years, and are now beginning to produce results. Our multidisciplinary accident investigation program, which is receiving international application through the CCMS, continues to produce the most comprehensive, in-depth studies of individual crash cases ever performed. Automated files built on thousands of cases are now being used to find answers to long-standing questions never before approachable. It is now possible, for example, to identify the specific causes of many classes of accidents in technical terms, rather than the often used vague generalizations such as "driver errors" or "vehicle failures." A recently completed study has quantified for the first time the specific contribution of different types of vehicle mechanical failures, either through wear out or breakage, to the total accident picture. This work is currently being expanded to include equal quantification of both driver and environmental factors according to a carefully designed taxonomy.

In addition to such in-depth studies, broad-based, statistical analyses are carried out on mass accident data files being developed as part of our Information and Data System. Just recently developed as part of this system is our Fatality Analysis File. This file consists of a detailed standardized report on *every* motor vehicle involved fatality occurring in all fifty States. The format and information coding has been carefully structured to permit rapid, low-error automation for computer analysis. Eventual expansion of this concept to a statistical sample of non-fatal accidents is being planned.

We also work closely with our counterparts in the Federal Highway Administration (FHWA) of DOT with respect to safety in the vehicle environment and of the vehicle/road interface. Recently, for example, we worked closely with FHWA in studying the problems associated with railroad grade crossing safety, including examination of locomotive crashworthiness alternatives, grade crossing signing, and the operational safety of wide buses. Our Office of Driver Performance also works closely with the highway human factors and road design engineers in the areas of bridge and guardrail design, performance requirements for frangible sign and overhead lighting supports, motorist information and warning system research, etc. We plan to jointly study the very important question of the relative cost/benefits associated with crash energy attenuation of highway systems, such as energy absorbing barriers and vehicle crash energy management systems. Each of these systems potentially involves billions of dollars in future public expenditures. The FHWA work in skid accident reduction through surface treatment of road surfaces and improved materials and methods of road surfacing is coordinated with our work in achieving improved tire traction characteristics. These are some of the examples of our involvement in research associated with the total traffic system.

It is, of course, impossible in these few minutes to cover in detail or even mention the wide range of research and development programs now underway within NHTSA. My purpose, then, has not been to present a comprehensive coverage of our R&D effort, but rather to illustrate by a few samples a fact that we all indeed recognize, namely, much more than the design and fabrication of ESV's will be required to advance highway safety. Our ESV program, which, in fact, accounts for about 20 percent of our total R&D budget in NHTSA, is not a separate, self-contained, all-encompassing project. It represents, rather, the synthesis and visible integration of a broad range of research efforts into composite vehicles that can set a pattern for production vehicles in the years to come.

I would now like to turn to the very important question of the relationship of our research program and rulemaking. As you probably know, NHTSA was created by two acts of Congress in 1966 — the Motor Vehicle and Traffic Safety Acts. This legislation charged DOT with the responsibility of reducing the number of traffic fatalities in the United States and empowered them to set certain standards to accomplish this end. Recognizing that the automobile-driver-environment system is a complex one that can easily defy ready analysis, these acts also authorized NHTSA to perform such research as necessary to support the rulemaking activity. The exact manner in which this support is provided varies according to a number of factors, including the relative state-of-the-art of the technology involved, the complexity of the problem and the priority of the problem. Also, much support is provided by our data operation in the fashion outlined above. The research program should, therefore, reflect the anticipated needs of the operating components of NHTSA.

The ESV program is a research program and as such it also supports rulemaking. It is, admittedly, a very visible and rather substantial portion of our research program. Since it is the only total vehicle systems program that we pursue, it is the obvious test bed for new ideas and innovative approaches that can be integrated in the accepted systems manner. In so doing, we hope to establish clearly and in an open forum what is feasible and within the state-of-the-art. This is the process by which this and other research programs contribute to the rulemaking process.

Ladies and Gentlemen, I would now like to take this opportunity to express to all of the participants in the International ESV Program the extreme gratitude of the United States for your industrious efforts to advance the cause of improving motor vehicle safety.

To our hosts, the Government of Japan, JAMA, MITI and MOT, I extend a special thanks for inviting us to your wonderful country, and for organizing a well-planned and productive conference in the beautiful city of Kyoto at this magnificent International Conference Center.

To all the governments and companies that have given so generously of scarce fiscal and engineering resources, I express special appreciation.

The progress that has been made, as is so evident from your presentations and the Experimental Safety Vehicles on display, is truly impressive. Those of you who have built complete ESV prototypes are deserving of special recognition. In our host country of Japan, I congratulate Toyota, Nissan, and Honda for their attractive and well-engineered vehicles and for the impressive scope and seriousness of their programs.

In Germany, the Daimler-Benz and Volkswagen ESV's speak for themselves. The development of the driving simulator and the progress made in addressing the serious problem of vehicle aggressiveness by VW is worthy of praise.

In Italy the impressive presentation by the Fiat Motor Company was certainly one of the highlights of the Conference. Deserving of special mention is their development of ESV's in several weight classes and work in vehicle handling and car-to-car compatibility. Anyone who has worked with the problem of mathematical modeling driver-vehicle behavior will understand the difficulty and must appreciate the work by Alfa Romeo.

FUTURE SAFETY STANDARDS AND THE ESV PROGRAM

SECTION FIVE PART 2

INTRODUCTION – COMMITTEE OF COMMON MARKET CONSTRUCTORS (CCMC)

M. OUIN, *Secretary General*
Renault

Mr. Chairman, I should first of all like to thank you and the NHTSA for inviting the CCMC to contribute to this conference.

I shall briefly outline what our organization is. The CCMC is an association of car manufacturers in the Common Market established between BLMC, Citroen, Daimler Benz, Fiat, Peugeot, Renault, and Volkswagen. Alfa Romeo and BMW subsequently joined us. This association of Common Market manufacturers was set up within the Common Market framework by manufacturers who want to make cars safer, while reducing the nuisance to the environment created by cars.

This has been done within the Common Market framework because the manufacturers involved feel that, thanks to the institutions of the European Economic Community and the Brussels body, it would be easier to obtain regulations that would affect vehicles in the future in as uniform way as possible so as to preserve free trade and the free exchange of cars within the Common Market. Of course they also hope that future regulations will be established within the Common Market so that they can be effective over as broad an area as possible.

The manufacturers involved aim at improving car safety as much as possible and as fast as possible. That is why within our association we decided to exchange all research information available on studies and tests carried out concerning car safety. I am merely referring to safety within the framework of this conference, of course. They have also undertaken to make available to governments both within the Common Market and the Brussels Commission all their experience, and scientific and technological information that will help to establish sound safety regulations on an economic and effective basis, taking into account the cost-benefit ratio and the special characteristics of European produced cars.

The CCMC is a recently established organization. It was set up only 3 months ago. As you know, and indeed as you have seen today, the members of the CCMC individually have all worked very hard to advance safety. They are only now beginning to exchange information on their respective work and they are trying to make this work available to the governments and to the public as broadly as possible, so that all the technical data that is gathered will be available. As an initial example and contribution of the work of this association to automobile safety, I am very happy to introduce the papers to be presented by the CCMC, on the one hand by Dr. Mackay, on the other hand by Dr. Tarriere. Dr. Mackay works at the University of Birmingham and also works for BMS, and it was in the framework of our association's presentation that he will talk to you about traffic accidents in Europe and studies related thereto. As to Dr. Tarriere, of the Renault-Peugeot Association, he will talk about the effectiveness of 3-point safety belts in real life accidents.

SECTION FIVE PART 2

FIELD STUDIES OF TRAFFIC ACCIDENTS IN EUROPE

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INTRODUCTION

For an experimental safety vehicle programme to be successful, it is imperative that the specifications for the design of advanced vehicles reflect the reality of accidents as they occur on the roads. It is therefore important for field studies of accidents to be undertaken so that the characteristics of the real world are understood. Only then is it possible to optimize the several conflicting requirements of protection to the various classes of road users in the great range of collision types and severities which occur in reality.

The Committee of Common Market Constructors has therefore adopted the policy of including field accident studies as an integral part of the ESV contribution. This paper reviews some of the general features of European accidents undertaken by the CCMC.

The aim of this paper is to outline the similarities and differences between Europe and North America, and also to begin to examine what detailed information is available on the frequencies and severities of accidents of different configurations, so that, eventually, appropriate specifications for ESV development can be optimized and refined.

THE GENERAL ACCIDENT SITUATION

Before considering car occupants, it is essential to examine the relative importance of accidents to all classes of road user. Table 1 presents some selected information on deaths and injuries in Europe and elsewhere. In general, for every fatality there are between 27 and 37 non-fatal injuries, depending on the environment and the reporting procedures. For car occupants, the ratio of fatally injured occupants to non-fatally injured occupants is 1 to 37.

The relative proportions of different classes of road user fatalities are shown in Table 2. In considering overall strategies, it is important to note that in Europe pedestrians and motorcyclists (including cyclists) together make up the majority of traffic fatalities (50.9%); car occupants only constitute 44.3%. This proportion is in marked contrast to the United States, where car occupants constitute 62.8% of the total.

Therefore, the consideration of pedestrian safety in particular must occupy a higher priority in Europe than in North America. Also of great importance are the large number of motorcyclists killed in Europe. Table 2 shows that Britain in particular has a relatively severe pedestrian situation, while Italy has a high incidence of motorcycle casualties. These facts should be borne in mind when considering national priorities.

CCMC SAMPLE STUDIES

For several years accident investigation teams have been collecting information in Europe. This research has been expanded in the last two years with the NATO programme of the Committee on the Challenges of Modern Society. The Committee of Common Market Constructors has contributed substantially to this field accident investigation programme, with the result that samples of accidents are

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now available for analysis. The data which follows come from studies conducted in France by Peugeot-Renault, in Germany by Volkswagen and Daimler-Benz, in Italy by Fiat and in Britain by the University of Birmingham.

Because each team is working in its own environment there are obvious differences between the samples. Also, each study is not necessarily representative of accidents in its own particular area, and one of the most important aspects of future ESV programmes should be the development of reliable and representative field accident data. However, although the sample of accidents collected by each team has its own internal and external biases, it is useful to examine certain basic collision characteristics.

Agreements were reached between the teams as to the methods used for recording the information and for analyzing the resultant data with common procedures. All of the teams have restricted their investigations to current model cars in injury-producing accidents.

The team of Peugeot-Renault concentrated on collisions occurring mainly on expressways. The Fiat and Volkswagen teams provided samples of collisions restricted to single impacts between cars and rigid objects. The Daimler-Benz team examined collisions in an environment containing higher proportions of autobahn mileage than the other samples. Therefore, the sample from the Continent of Europe represents much more severe collision types than the total population of injury-producing accidents. The British sample, also restricted to occupant injury in current model cars, is likewise biased towards the more serious end of the accident spectrum because of the selection procedure used. This emphasis on serious fatal injuries is demonstrated in Table 3, which shows the ratios of fatally injured car occupants to non-fatally injured car occupants for the samples of accidents from the five teams, in comparison to the overall ratios established for all car occupant injury accidents in each country.

[illegible]

Therefore, the following analysis should not be considered as representing typical accident conditions. The analysis, in fact, represents the upper boundaries of collision severity; further carefully structured field accident studies will be needed to evaluate typical conditions of actual accidents.

COLLISION SEVERITY

The total sample of accidents for the five teams represents 986 cars in collisions producing injury to at least one occupant. The types of accident are outlined in Table 4. Some 80% of the cars were involved in single impact collisions. The remaining 20% included multiple impacts, rollovers and combinations of these events.

The impacts to each car were then classified, where possible into the most appropriate Equivalent Test category (1). This was done by examining the damage to the vehicle and relating that damage to the damage which would have been produced in the closest equivalent test of the ESV specification. The results of that analysis are given in Table 5, which shows the frequency of the various collision types.

Of particular importance is the relatively low frequency of the distributed frontal type of collision. Front angled impacts are twice as frequent as the distributed type.

[illegible]

[illegible]

Over one third of impacts were found to be unclassifiable into any Equivalent Test category. A large number of these are of the under-run type, involving an impact with a large commercial vehicle. It is known that such collisions are particularly severe. In Britain, for example, a sample study suggests that some 50% of car occupant fatalities occur in car/truck collisions (2). Better crash performance cannot be achieved here by the control of the car structure, but must be realized through improvements to commercial vehicles.

The low frequency of pole impacts, particularly in side collisions, suggests that such a collision type is not of great importance overall.

INJURY SEVERITY AND EQUIVALENT TESTS

Having classified the cars into the several Equivalent Test categories, each car was ranked by collision severity in terms of its Equivalent Test Speed (E.T.S.). This was done on the basis of the extent of the damage to the car in comparison to the known damage which would occur to the same make and model of car in an Equivalent Test. Such an assessment is therefore a comparison based on the energy involved in a collision. For frontal impacts the damage was related to fixed barrier collisions, while for rear and side impacts each car was related back to the equivalent damage produced by a mobile barrier (1).

Such assessments of equivalent barrier speeds have been used in the past (1,4) although it has been acknowledged that such a procedure is necessarily approximate. The CCMC has a study under way which estimates collision severity based on the velocity change (ΔV) experienced by each vehicle in a collision. This procedure allows the accident severity therefore to be assessed in terms of the velocity change experienced by the occupant rather than the car impact speed. This method of analyzing accident data has not been used in this paper, but work is under way and results will be presented in the near future.

The overall severity of injury to each front seat occupant was assessed using the Abbreviated Injury Scale developed by States (3). Table 6 illustrates occupant injury severity frequency by the major types of Equivalent Test. This table demonstrates that fatal injuries are over-represented in this sample of accidents in comparison with the overall ratio of fatal to non-fatal occupant injuries for national accident figures.

Having therefore a measure of occupant injury severity and collision severity, an analysis is given below of the Equivalent Test Speeds for various levels of injury severity on an occupant basis.

[illegible]

FRONTAL IMPACTS

The three types of frontal impact are grouped together and shown by various levels of injury in Figure 1. These curves show the E.T.S. values for the European data. For moderate injury levels (A.I.S. values 02 and 03) 80% of collisions occur at less than 22 mph, and for severe and fatal injuries 80% of collisions occur at 40 mph and less.

It is of interest to compare the speed distribution for these European data with speeds described for

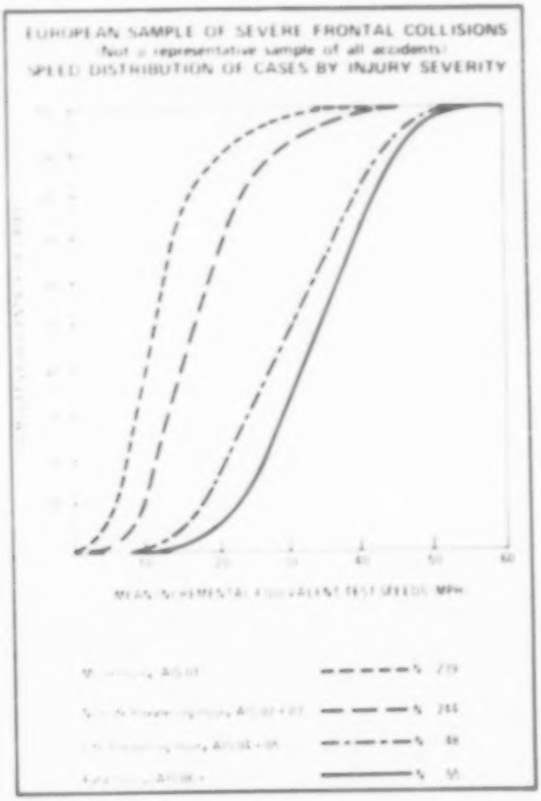

Figure 1

accidents in the United States. This has been done for the non-fatal cases in Figure 2. In the present sample there were only 55 fatal cases, which, considering the biases present in the sample, represents too small a number for comparison.

Figure 2 shows the cumulative frequency curve for the non-fatal European cases, together with 95% confidence limits based on the sample size of 531 cases. This means, for example, that there is only a one in twenty chance that the upper impact speed limit for 80% of these collisions will be outside the range of 20 to 23 mph.

Also plotted on Figure 2 is a curve for injury producing frontal accidents in the United States quoted by Carter (4). It is evident from this information that there are considerable differences between the two continents if these data represent in any way the typical circumstances.

The data on which the Carter curve is based come from an analysis of ACIR accidents (5) which are mainly rural, probably not typical of the total U.S. environment. The crash severity of the original accident cases was assessed in terms of the estimated travelling speed of the cars involved. These estimates were based on posted speed limits, witnesses' statements, prevailing traffic speeds and the vehicle damage.

The derivation of the Carter curve involved classifying the ACIR data into 20 mph increments of estimated travelling speed values with the further assumption that the impact speeds were 10 mph less than the travelling speeds (6). For the speed range involved this means that the curve is largely defined on the basis of three points only, and should not, therefore, be considered as an accurate description of the equivalent test speeds in frontal impacts in the United States.

SIDE IMPACTS

The equivalent test speeds for side impacts have been evaluated for all injury severities as a single group because of the small numbers involved. The analysis was restricted to those occupants seated on the struck side of the vehicles. Figure 3 shows these data, and demonstrates that 80% of side impact equivalent test speeds were below 25 mph. Only 10 of these side impact cases involved fatally injured occupants and so no separate analysis has been performed.

REAR IMPACTS

Rear end collisions have been treated in a similar manner to give a cumulative speed distribution shown in Figure 4. In the entire European sample there were only 34 cases in total, of which 28 were minor injuries, two were moderate and two were fatalities. The curve shows that only minor injuries are being sustained in relatively high speed rear end collisions, with 80% of collisions occurring at E.T.S. values of 32 mph and less. The absence of any reasonable

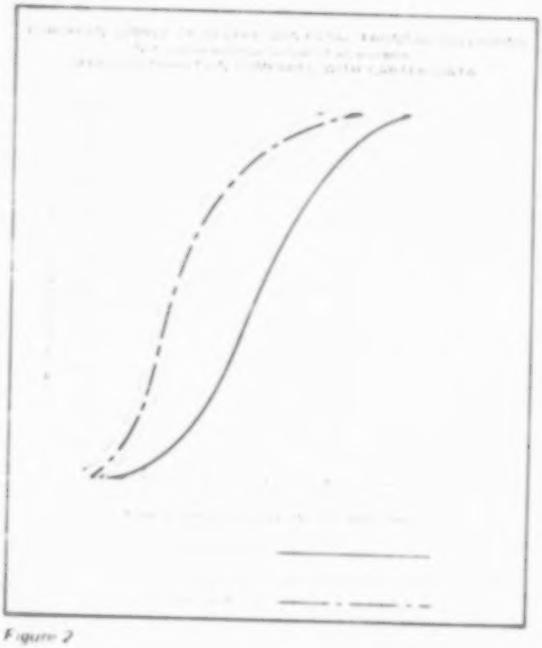

Figure 2

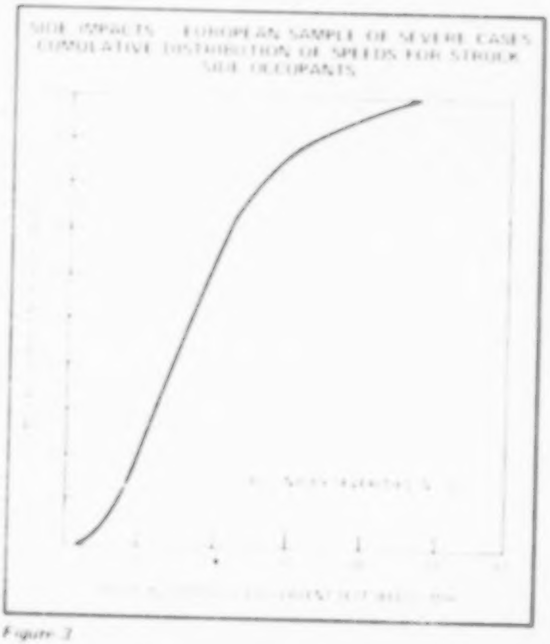

Figure 3

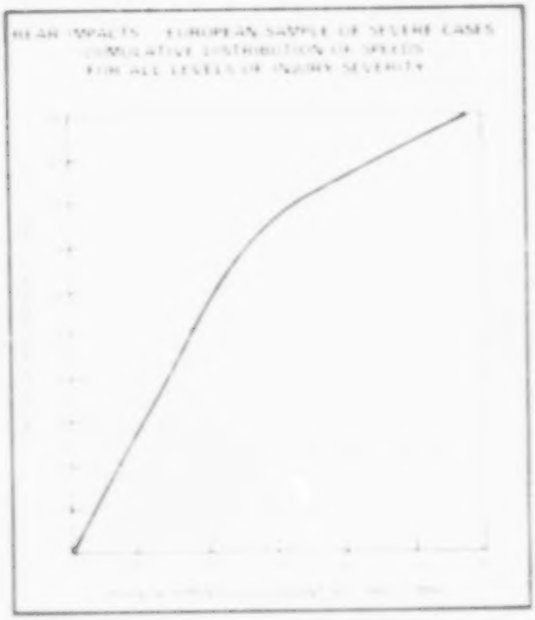

Figure 4

number of serious and fatal injuries and the fact that only 4.3% of all collisions involve rear impacts, suggest that the likely benefits obtained from improved rear end performance will not be great. Careful cost-benefit assessments should certainly be attempted before establishing higher performance requirements.

CONCLUSIONS

This paper has illustrated how field accident studies can be used as an aid to defining appropriate specifications for crash performance. Because of the complex nature of the reality of accidents on the roads, the data quoted in this study are not sufficiently representative to allow detailed conclusions to be drawn. In particular, the sample represents only the severe end of the accident spectrum. However, certain general points should be made.

1. Pedestrian and motorcyclist collisions occupy positions of much greater importance in Europe than in North America. Programmes to ameliorate pedestrian trauma are likely to have greater benefits than a single-minded pursuit of occupant protection at higher and higher impact speeds.
2. In view of the differences in the traffic environments between North America and Europe, it is likely that crash conditions for cars vary considerably; these data collected by the CCMC suggest that such is the case.
3. The European data based on this sample from the severe end of the accident spectrum, show that some 98% of injury-producing frontal impacts occur at Equivalent Test Speeds of less than 40 mph. Also, some 84% of critical injuries and some 73% of fatal injuries occur at less than that speed. Insufficient data are available on fatal cases to specify likely benefits of improved crash performance, but the high frequency of car/truck cases (unclassifiable on an E.T.S. basis), suggest that improved collision performance above 40 mph in a barrier test is unlikely to yield great benefits, and will necessarily compromise other crash performance requirements. Therefore, 40 mph should be considered an appropriate goal for frontal impacts in ESV specifications. That figure may be refined further in the future as better accident data become available.
4. For side impacts it is clear from previous work (1) that the car-to-car situation, simulated by a shaped barrier, represents the dominant accident type. Pole impacts occur with relatively low frequency. The major problem with side collisions is to optimize the front and side structures of vehicles together in order to match the front and side structure characteristics (7).
5. Rear impacts in Europe represent a small part of the accident population, particularly where fatal injuries are concerned. The likely benefits to be gained from improved rear end performance require careful cost benefit evaluation.
6. Accident investigation must be an integral part of any ESV programme. Initially, the reality of accidents in different parts of the world must be studied in order to establish national priorities which will vary around the world. As design changes are introduced into vehicles, it is also necessary to conduct field accident studies in order to monitor the effect of such changes on the injuries to road users in actual collisions. Such investigations provide the final judgment on any design change.

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FUTURE SAFETY STANDARDS AND THE ESV PROGRAM

SECTION **FIVE** PART **2**

EFFICIENCY OF THE 3-POINT BELT IN REAL ACCIDENTS

DR. CLAUDE TARRIERE, *Doctor of Medicine*
Laboratoire de Physiologie et de Biomecanique
Association Peugeot Renault

The study we are presenting compares the severity of injuries observed on vehicle occupants according to whether or not they wore a safety belt, the violence of impact being equivalent.

Medical as well as technical data were collected by three bidisciplinary teams:

- The Impact Laboratory of the National Organism Road Safety (Onser - Lyon - Bron)
- The Accident Research Unit of Birmingham University
- The Laboratory of Physiology and Biomechanics from the Peugeot-Renault Association.

One hundred and sixty cases of occupants wearing 3-point belts (pelvic + torso restraints) were compared to 782 cases of occupants not wearing safety belts.

For the first time, the evaluation of collision violence, particularly in vehicle-against-vehicle impacts, used evaluations of velocity changes of vehicles according to the method which was presented to you by C. Prost-Dame.

For each type of accident (frontal collision, side collision and rollover), the general balance-sheet is presented in synthetic tables completed by numerous clinical studies of cases which are either very representative or very particular (see Tables I through III).

Efficiency Of Belts In Frontal Collisions

Figure 1 shows the distribution of collisions according to how the impacted zone of the vehicle is distributed or localized, symmetrical or asymetrical.

TABLE I
DISTRIBUTION OF THE OCCUPANTS BY TYPE OF ACCIDENTS

SEAT BELT		ACCIDENT			
		FRONTAL	LATERAL		TURNING OVER
		[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE II
SEATED POSITION AND SEX OF THE OCCUPANTS

		MEN		WOMEN	
		SEAT BELT		SEAT BELT	
		WITH	WITHOUT	WITH	WITHOUT
DRIVER	Number	68	617	0	97
	%	17.6	[illegible]	0	18.7
FRONT SEAT PASSENGER	Number	11	58	26	211
	%	[illegible]	76.4	8.1	44

TABLE III
DISTRIBUTION ACCORDING TO THE AGE OF THE OCCUPANTS

	SEAT BELT	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	UNKNOWN	
DRIVER	WITH	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Number
		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	%
	WITHOUT	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Number
		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	%
FRONT SEAT PASSENGER	WITH	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Number
		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	%
	WITHOUT	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Number
		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	%

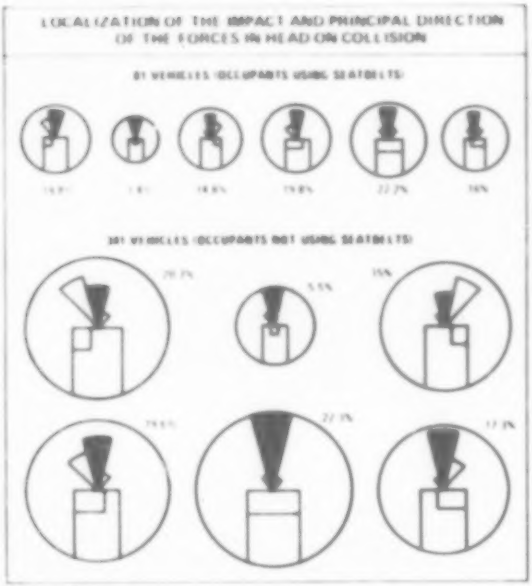

LOCALIZATION OF THE IMPACT AND PRINCIPAL DIRECTION OF THE FORCES IN HEAD ON COLLISION
81 VEHICLES (OCCUPANTS USING SEATBELTS)
18.8%
19.8%
22.7%
16%
301 VEHICLES (OCCUPANTS NOT USING SEATBELTS)
20.7%
5.5%
15%
27.3%
17.3%

Figure 1

The same distribution can be observed *with* and *without* belts and two impacts out of three are offset.

The main direction of deceleration forces is also indicated in Figure 1. The area of each triangular zone is proportional to the number of cases corresponding to this direction. If most of the impacts are axial, the frequency of obliquely directed impacts is very high, particularly in the sample without a belt.

The violence of the impact was estimated in terms of *speed variation during the period of vehicle deformation* (ΔV).

Figure 2 shows the cumulative histogram of impact violences in frontal collisions. One observes that occupants wearing belts were involved in collisions having a severity much higher than those of the non-belted occupants. For half of the most severe collisions the violence of impact with belts exceeded on an average that of impacts without belts from 10 to 15 km/h. This difference is significant at 0.01.

Figure 2 shows the confidence bands around the histogram of the cases without belts. On the average, the more violent cases with belts can be partly explained by the fact that a certain percentage of moderately violent accidents whose belted occupants were not sent to a hospital (uninjured) were overlooked by the investigators (accidents with one single vehicle in particular). We stress the fact that the real efficiency of belts is for these reasons underestimated in this study.

General Balance Sheet

Six hundred and fifty-eight occupants, all of them sitting on front seats and involved in frontal, injury-producing collisions, are presented in Figure 3, according to the severity of the injuries and the collision violence:

- 118 occupants *with* belts are represented by *triangles*.
- 540 occupants *without* belts are represented by *open circles*. Among the latter, the *ejected* (37 or 6.8%) are represented by *solid circles*.

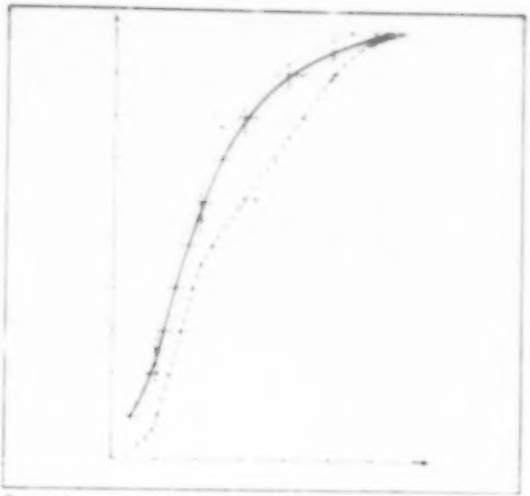


Figure 2

In the aggregate, a first rapid study of the Figure allows one to note that the occupants *with* belts are situated, in relative frequency, much lower in the range of injury severity for each impact violence. This situation characterizes graphically the quality of the protection granted by the belt; it is all the better as the collision violence is more severe.

Applying a test of difference for each type of violence, as may be observed in the average curve, a substantially lower seriousness for the belted occupants (between 0.05 and 0.001) is noted, except for the range of violence from 36 km/h to 45 km/h where the difference is not significant.

Moreover, a test was applied to the 85 occupants involved in collisions where the speed variation (ΔV) was between 46 and 75 km/h. These are the cases corresponding to the right three columns of Figure 3, and for which the cases with and without belts are distributed homogeneously in the three classes of greater violence.

This test reveals a significant difference (0.04) of seriousness (equal or higher seriousness) at I.S.I. 4 for non-belted occupants and lower for belted occupants. This result is of great importance and must be underlined, for it proves that the limit of belt efficiency is higher than normally considered. Here, the belt reduces the injury severity up to the ΔV range of 66 to 75 km/h (1 to 2% of the frontal collisions in our sample reach such a violence).

The most interesting observations regarding this sample are: For collisions at less than 55 km/h (ΔV) no belted occupants died. As regards non-belted occupants, 72% of the fatal injuries (23 out of 32) occurred at a level of violence lower than the preceding one and the first of them occurred in a collision at a ΔV lower than 25 km/h (by ejection) and the two following ones in a collision at a ΔV between 26 and 35 km/h (without ejection). When ΔV is lower than 55 km/h, the probability of being killed or severely injured is six times lower with a belt. Ninety-six percent of the frontal collisions (with injury requiring hospitalization of at least one occupant) occurred at a violence lower than that corresponding to such a ΔV.

For the most severe impact violences (ΔV between 56 and 75 km/h), the two less severely injured occupants wore belts (Case G will be analyzed later). The probability of being killed or seriously injured is still *twice as low with* a belt in collisions of exceptional violence.

The five cases of the most seriously injured occupants wearing a belt (four people killed and one very seriously injured) are all special cases observed for an exceptional collision violence:

- Two killed on the spot (I.S.L. 8) by intrusion of the obstacle into the occupant compartment (Cases A and B). The belt was insufficient and even useless under the circumstances.
- Two deaths (Cases C and D) within the first hours (I.S.L. 7) and one injured (I.S.L. 6) (Case E) occurred when non-belted rear-sitting occupants were projected forward causing the seriousness of the injuries, besides the overloading thus exerted on the belt resulting in belt breakage (Cases D and E). These special circumstances justify the fact that we do not merely proceed to a statistical analysis of the sample. The clinical analysis of some cases is an essential complementary approach.

Thirty-seven non-belted occupants were ejected; among them, six were dead (I.S.L. [illegible]) and seven seriously injured, which still points out the severity of these ejections. *The [illegible] complete protection against ejection.*

The comparison between drivers and [illegible] occupants [illegible] with or without belts (Figures 4 [illegible]) show [illegible] the [illegible] (Figure 6).

ANALYSIS OF CASES INDICATED A TO I IN FIGURE 3

Case A And B

Simultaneous [illegible] the [illegible] of the occupant compartment [illegible] Cases A and B.

These [illegible] when the [illegible] compartment [illegible] which [illegible]

The [illegible] optimizing the vehicle structure. A [illegible] that it is possible to [illegible] the structure so that the complete destruction of half the occupant compartment is avoided, as shown in Figures 5a and b, and that a restraint system can be adapted which would avoid severe impact against the sides, are we sure that the restraint constraints would still be tolerable? This is one of the basic questions clearly introduced in the Fiat presentation, for which there is no clear answer. Yet, we must not forget that these questions apply to limited cases which in Figure 2 represent only 2 to 4% of the frontal collisions in our total sample.

Cases C, D, And E

Front belted occupant may be compromised by the presence of a rear non-belted occupant.

Case C

A Renault 16 was impacted on the left front by a Simca-Beaulieu running in the opposite direction. The ΔV was between 56 and 65 km/h.

Occupants

- Two front-seat occupants wearing 3-point Association belts.
- Two rear occupants without belts.

Medical Facts

- The driver (33 years old, 1.90 m, 85 kg)
  - No injury of head or cervical spine
  - Simple thoracic contusions
  - Fracture of cotyle rear wall and right femoral head with entirely recovered sciatic nerve lesion

  Severity Index of Lesions: 3
- The front female occupant (33 years old)
  - Skull trauma, loss of consciousness
  - Multiple rib fractures with displacement
  - Sternum fracture
  - Thigh-bone open fracture

  Severity Index of Lesions: [illegible]

Comments

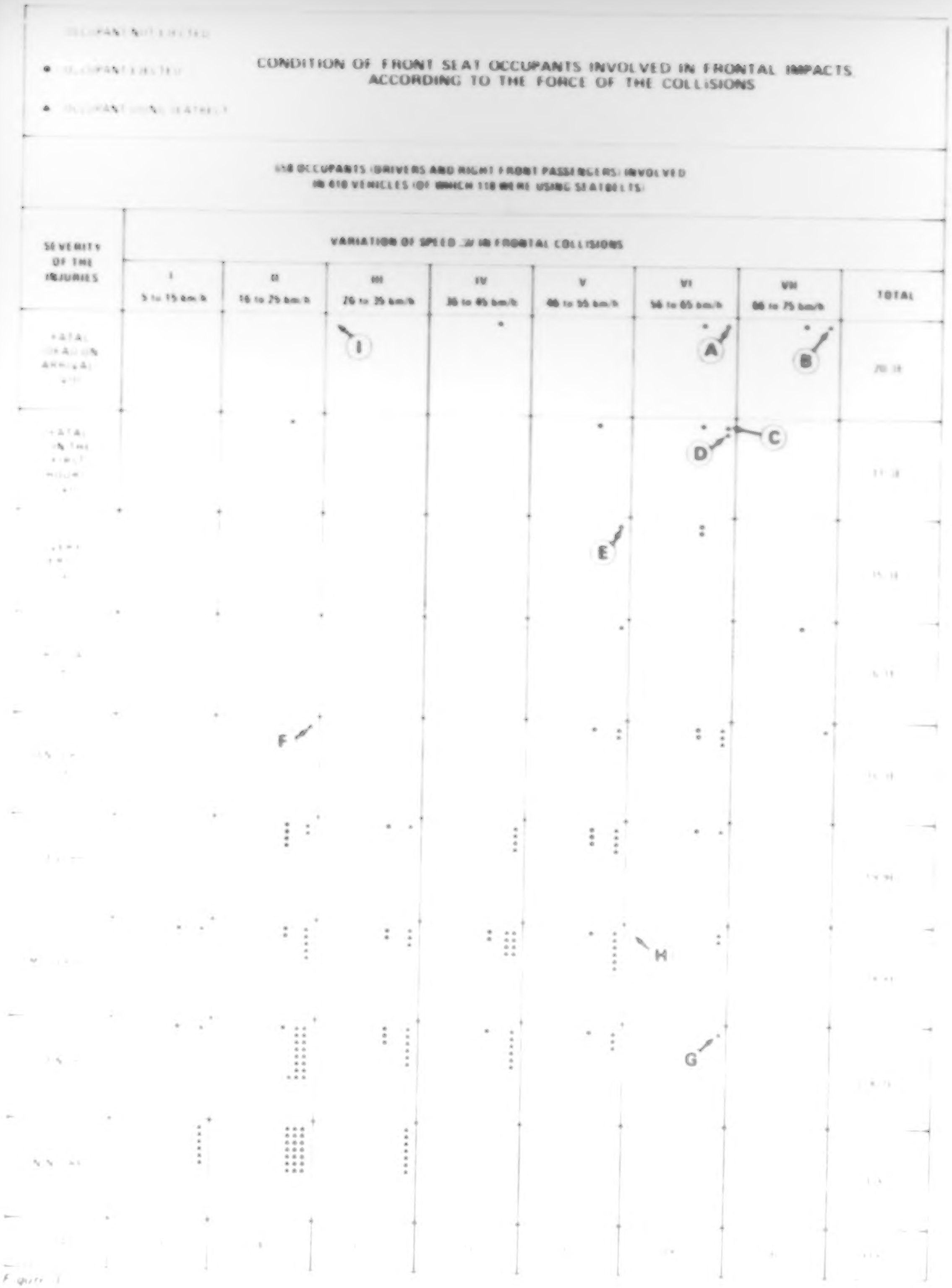
As can be seen in Figures 6a and b, the occupant compartment, though showing a certain reduction in volume, may be considered as compatible with the survival of the front female occupant.

The belt should have shown a [illegible] play but only two straps were broken while five straps of the driver's belt were broken despite the steering wheel and the oblique nature of impact [illegible] the right front part.

As a matter of fact, the [illegible] front female occupant was the projection of the [illegible] 85 kg male who was in the rear. He [illegible] the seat out of its guides, crushed the [illegible] against the dashboard and [illegible] head [illegible] hyperflexion (cervical fractures). [illegible] the male was ejected through the windshield and [illegible] to a violent skull trauma when he struck the [illegible] with fatal effects.

Case D

A Ford Capri was involved in a [illegible] collision with a much heavier [illegible] ΔV [illegible] km/h.



OCCUPANT NOT EJECTED
OCCUPANT EJECTED
OCCUPANT USING SEATBELT
CONDITION OF FRONT SEAT OCCUPANTS INVOLVED IN FRONTAL IMPACTS ACCORDING TO THE FORCE OF THE COLLISIONS
OCCUPANTS (DRIVERS AND RIGHT FRONT PASSENGERS) INVOLVED IN VEHICLES (OF WHICH 118 WERE USING SEATBELTS)
SEVERITY OF THE INJURIES
VARIATION OF SPEED ΔV IN FRONTAL COLLISIONS
I 5 to 15 km/h
II 16 to 25 km/h
III 26 to 35 km/h
IV 36 to 45 km/h
V 46 to 55 km/h
VI 56 to 65 km/h
VII 66 to 75 km/h
TOTAL
FATAL DEAD ON ARRIVAL
FATAL IN THE FIRST HOURS
I
A
B
C
D
E
F
G
H

Figure 1

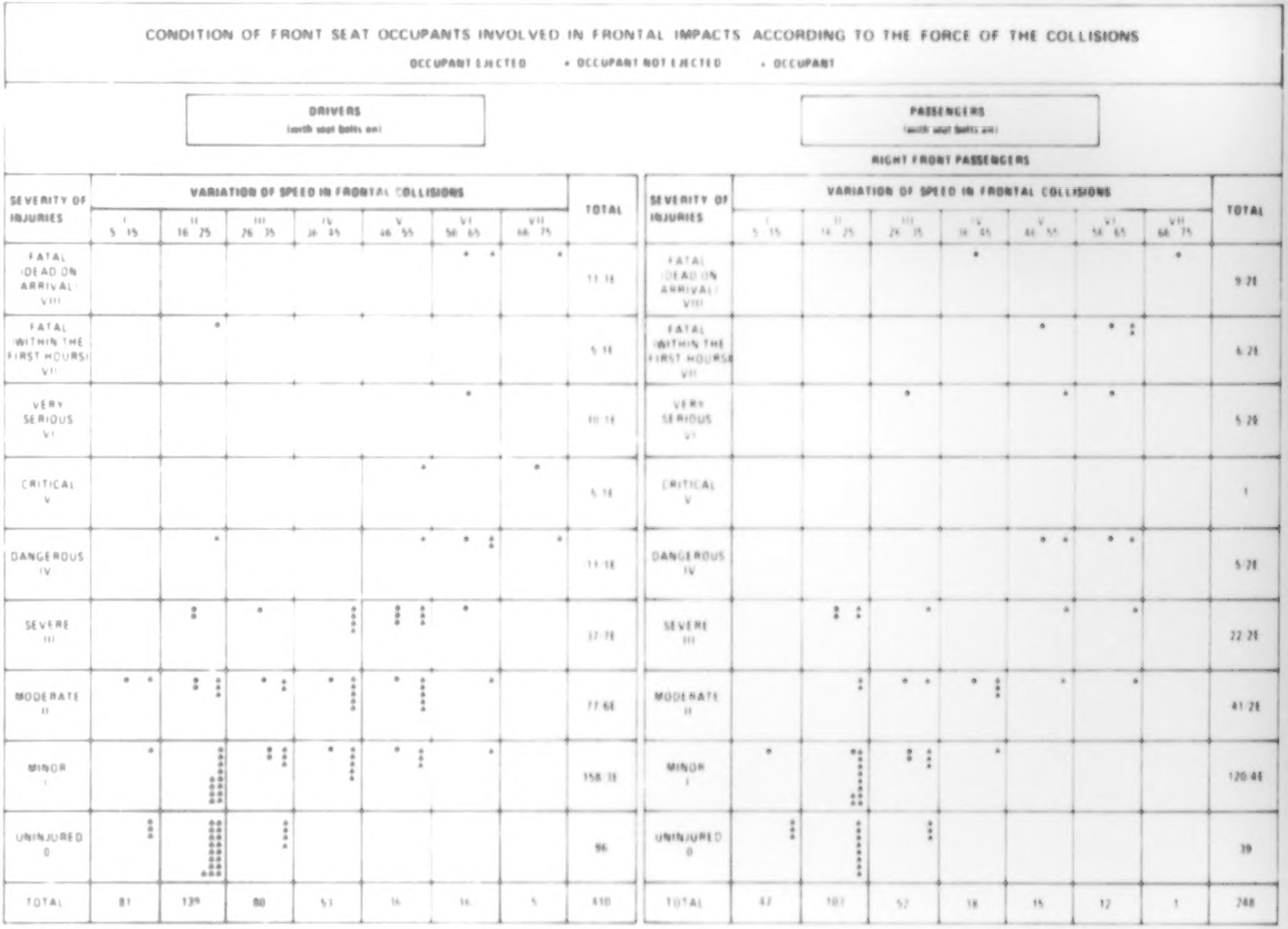

CONDITION OF FRONT SEAT OCCUPANTS INVOLVED IN FRONTAL IMPACTS ACCORDING TO THE FORCE OF THE COLLISIONS

Figure 4

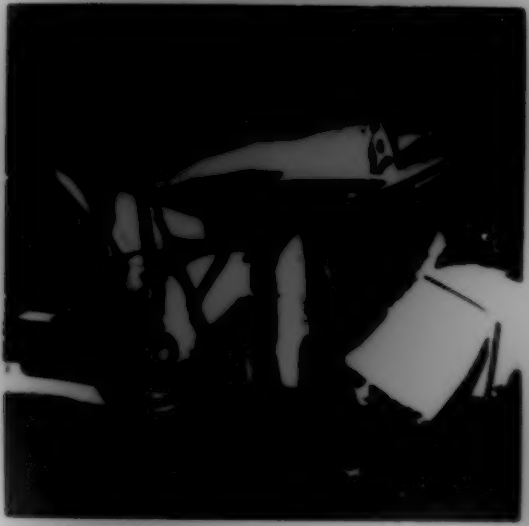


Figure 5a

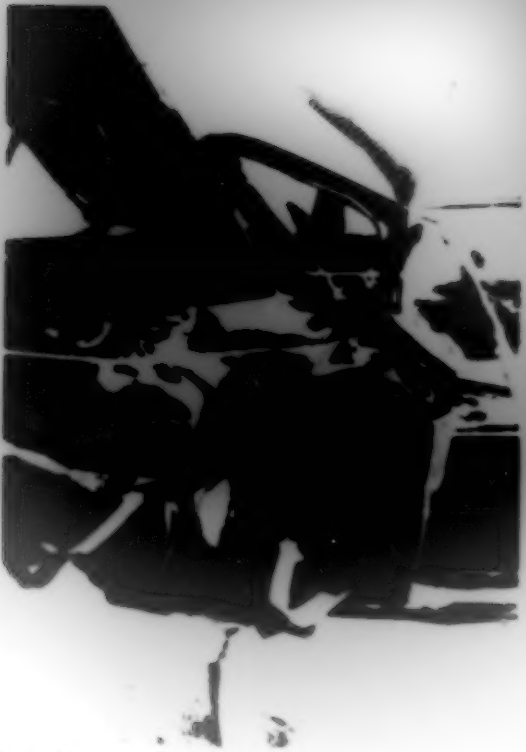


Figure 5b

Occupants

- 51-year-old female occupant (1.63 m) restrained by a belt
- 40 kg dog [illegible] in the rear seat

Figure 6a

Figure 6b

Medical Facts

At impact, the dog was projected against the front seat back causing a substantial deformation of the seat. The belt was subjected to an overload and gave way.

- The female occupant did not survive her injuries. Severity Index of Lesions: (not indicated).

Case E

A Simca 1501 impacted with its right front structure, the left rear structure of a lorry and then, during overtaking, the latter struck an oncoming 404

The main impact was slightly offset, which resulted in a speed variation of about 50 km/h.

The front male occupant of the Simca (33 years old) wore a belt which was broken at the outer floor anchorage level under the push of a non-belted rear occupant (Figures 7a, b and c).

Medical Facts:

- Loss of consciousness for over 24 hours.
- Fractured face.

Figure 7a

Figure 7b

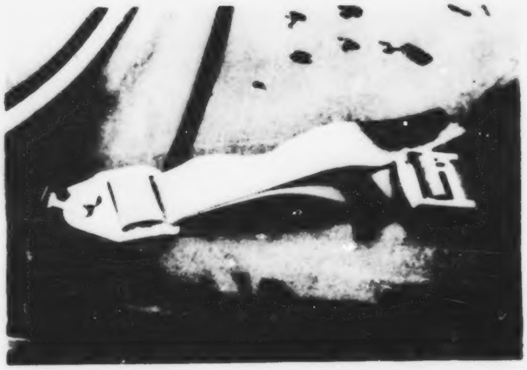


Figure 7c

- Fractured lower members.
- A certain degree of non-specified definitive disablement.

Severity Index of Lesions: 6.

The 29 year-old restrained Simca driver sustained a slight skull trauma and lacerations of the face (among which were eye injuries).

Severity Index of Lesions: 2.

As concerns the adverse vehicle (Peugeot 404).

Medical Facts:

- Front occupant was dead.
- Driver sustained multiple rib fractures with displacement and remained disabled during more than three months.

Severity Index of Lesions: 4.

Comment:

Cases C, D, and E point out the importance of restraints for rear occupants for their own protection as well as for the protection of front seat occupants. Improvement of the front seat compartments (e.g. stiffening of the seat and its floor anchorage) and their design (e.g. a strongly-built headrest) might in some cases protect the front occupant. It should be noted, however, that such solutions are much more expensive than the "3-point" or "pelvic" belts used for rear occupants.

Case F

This case concerns a Fiat 128 which, following a traffic slow-down on a superhighway, impacted the rear structure of a Mercedes 220 which had just stopped. The impact speed was estimated at 40/50 km/h and it must have dropped down to about 25 km/h (Figures 8a, b, and c). Both the driver and the front occupant were restrained by Britix 3-point belts. Nobody was sitting on the rear seats.

Medical Facts:

- 61 year-old male driver (1.59 m, 66 kg):
  - sustained skull trauma without loss of consciousness, together with
  - bilateral frontal injuries,
  - left multiple rib fractures with displacement, and
  - fracture of left tibial plate.

Duration of hospital treatment: 11 days; disablement: two months.

Severity Index of Lesions: 4.

- 67 year-old female occupant (1.58 m, 59 kg):
  - sustained skull trauma with loss of consciousness, together with

- multiple lacerations of scalp and face (with traces of impact against windshield).
- fractures of left and right radius and cubitus.
  Severity Index of Lesions: 3.

Comment

These occupants struck the steering wheel (traces of driver impact, which account for rib fractures), the dashboard (view from inside of Figure 8) and probably the windshield. Displacements were not in harmony with the violence of the collision which leads us to believe that the belt straps were too loose.

In view of the moderate violence of the collision, these impacts might have been avoided with correctly worn belts. In any event, contusions probably would have been minor ones.

Case G

This is a case where a retractor belt gave excellent results. A Renault 16, with four restrained occupants, struck the side of a Hillman-Minx Sedan at approximately 90°. For the Renault the ΔV was about 58 km/h.

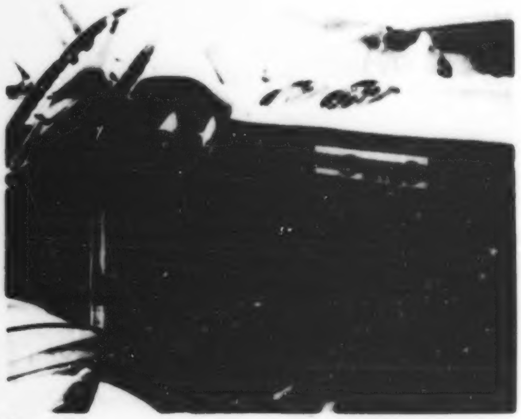


Figure 8a

Figure 8b

Figure 8c

Medical Facts:

- The 32 year-old male driver (70 kg, 1.72 m), was restrained by a 3-point retractor belt with inertia locking. He sustained minor lacerations of the face (caused by the windshield), and thoracic and knee contusions.
  Severity Index of Lesions: 1.

Comments:

This case illustrates a particularly favorable lesion-to-violence ratio for the impact. When compared with the preceding case, this example suggests that, from a statistical point of view, retractor belts will give better results for users than conventional belts, which are often worn too loose.

Case H

This is an analysis of the most favorable "unrestrained" case involving a slightly offset frontal collision between two Renault 8 vehicles. One vehicle underwent two more secondary impacts, after the main impact. ΔV was estimated at between 55 and 60 km/h.

The unrestrained 20 year-old male driver (1.80 m, 67 kg) was alone and sustained only a skull traumatism with loss of consciousness and some minor injuries. The steering wheel was substantially shifted into the occupant compartment, the rim being only slightly distorted.

Severity Index of Lesions: 2.

Comment:

One is entitled to think that a certain coupling took place between the driver and the foremost wall of the occupant compartment during a relatively long period of deceleration, since both Renault 8's (rear engine vehicles) showed a high capacity for normal distortion which was further increased in this accident by a substantial intrusion into the passenger compartment (Figures 9a, b, and c).

This explanation also applies to the occupants of the adverse vehicle:

Severity Index of Lesions for the driver: 3.

Severity Index of Lesions for the occupant: 6.

Comment:

Clearly marked traces of impact under the dashboard showed that the ejection was secondary relative to the projection into the foremost walls. This case is one of the good available illustrations of the non-significant idea of the so-called survival space when there is no restraint. (We prefer to speak of minimal space for restrained occupant deceleration).

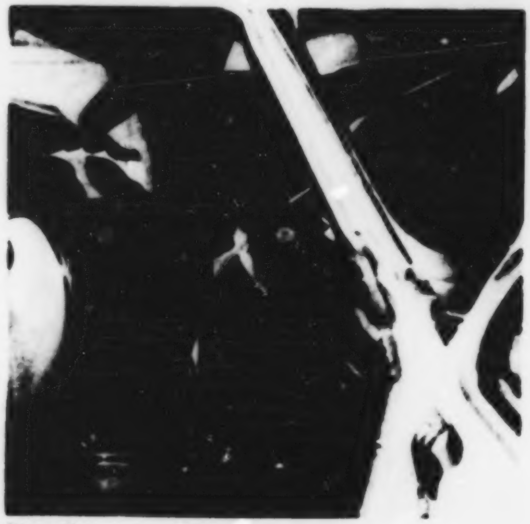


Figure 9a

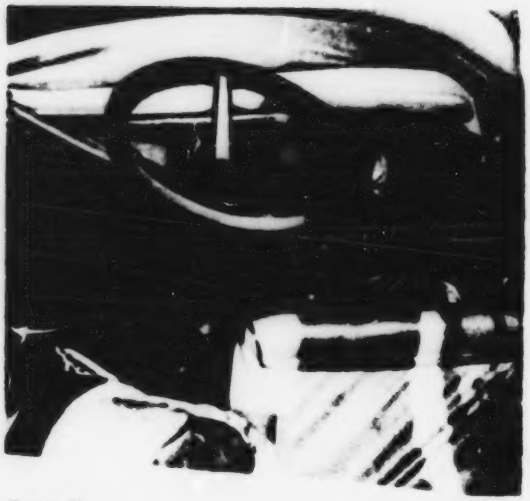


Figure 9b

Figure 9c

Case I

This was considered the most favorable "unrestrained" case without ejection.

A Triumph 2500 P.I. struck an oncoming Simca 1500. Maximum speed variation of the Triumph was approximately 30 km/h (ΔV). This was a well-centered impact without off-setting of either car (Figures 10a and b).

Medical Facts:

- Unbelted female driver of Triumph killed, sustained facial injuries; autopsy revealed aorta rupture. Other occupants in both vehicles only slightly injured without loss of consciousness, which seems normal in this type of collision.

Figure 10a

Figure 10b

Figure 10c

Comment:

A violent impact against the steering-wheel hub is certain. The three planar slightly rigid spokes of the steering-wheel were turned upside down and the hub projected outside. The spokes folded without substantial resistance and the thorax of the female driver hit the hub without substantial deceleration relative to the initial speed. In this moderately violent collision, a belt would have saved the life of the female driver.

Belted And Unbelted Occupants Inside Same Vehicle Involved In Frontal Collision

Such accidents make it possible to draw a comparison whose significance is very easy to determine since, during frontal collisions, the violence is the same for both front seat occupants. Eleven couples "driver/front occupant," with only one of each couple being restrained by a belt, were picked from our sample (Table IV).

It may be observed that for all cases, however violent the impact was, the "belted" occupant sustained less severe injuries than his "unbelted" neighbor.

Differences in seriousness are sometimes dramatic. Thus, in one moderately violent collision, the unbelted occupant was ejected and killed, whereas the belted occupant suffered only minor injuries. Another case was still more demonstrative since none of the two occupants was ejected.

TABLE IV

OCCUPANTS WITH AND WITHOUT SEATBELT INVOLVED IN THE SAME ACCIDENT

NUMBER OF DOCUMENT	FORCE OF COLLISION [illegible]	SEVERITY OF THE INJURIES	
		DRIVER WITH SEATBELT	RIGHT FRONT PASSENGER WITHOUT SEATBELT
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
		RIGHT FRONT PASSENGER WITH SEATBELT	DRIVER WITHOUT SEATBELT
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]

Case J

A Renault 12 Gordini with two occupants went out of control and struck a large tree. The impact was well centered on the front part of vehicle structure. It was a very violent impact, taking into account the nature of the obstacle; ΔV was estimated to be around 60 to 70 km/h (only 2 to 4% of frontal collisions have a violence equal or superior).

Medical Facts:

- The unbelted driver was killed on the spot, suffering polytraumatism and vertebral dislocation.
- Right front occupant was restrained and sustained injuries despite occupant compartment intrusion, which limited deceleration volume in the belt (only three strands were broken in the belt). The occupant did not show major injuries:
  - facial contusions and lacerations.
  - nose fracture.
  - hip contusions with slight crushing of femoral epiphysis.
  - fracture of left knee-cap without displacement.
  - fracture of the extremities of 2nd, 3rd, and 4th metatarsals of left foot I.S.L.2 - 15 days - hospital treatment.

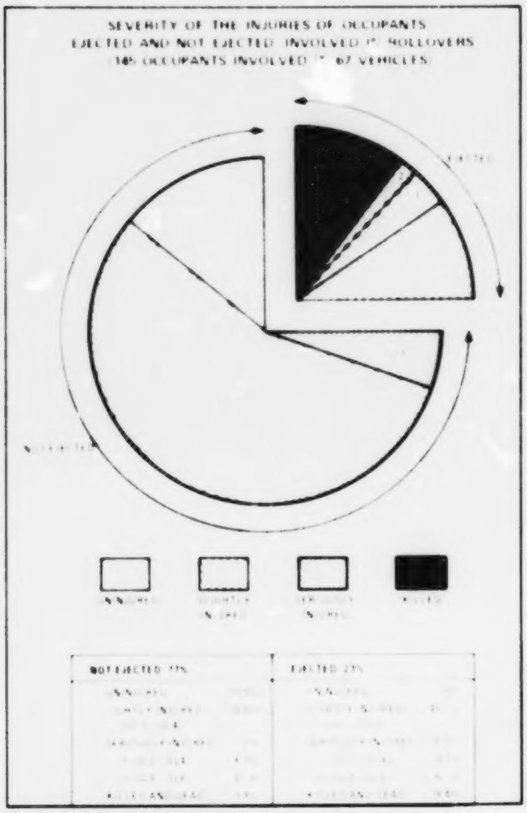


Figure 11

Efficiency Of Belts In Roll-overs

One hundred and forty-five occupants were involved in roll-overs, among them 14 with a belt.

The distribution of these occupants relative to the Severity Index of Lesions is presented in Table V.

On account of the limited sample with a belt it is impossible to test the statistical significance of the protection brought by the belt in this type of accident. Yet, it is important to draw attention to the fact that the seriousness of these accidents comes from ejection: 23% of the occupants were ejected and 40% of them were killed (Figure 11). Thus, Table V shows that 15 out of 16 occupants without a belt who present a Severity Index of Lesions ranging from 5 to 8 are ejected and that 13 out of the 15 killed were ejected.

Since the belt is totally efficient with respect to ejection (no ejections among the 160 occupants with a belt), we may thus consider that it constitutes a particularly efficient protection in roll-over. Moreover, we may state that of the 112 persons who remained in the occupant compartment during the 67 roll-overs analyzed, only two were killed. One of them wore a belt. It has to be noted that it is the only serious case with a belt, although an analysis of this case showed that the circumstances were very special (Case J) for it was a roll-over on the extreme part of an enclosing pole.

TABLE V

CASES OF CARS TURNING OVER

INDEX OF SEVERITY OF THE LESIONS	SEATBELTS WITH (14)	SEATBELTS WITHOUT (131)
0	4	19
1	4	64 of which 5 were ejected
2	1	23 of which 10 were ejected
3	2	6 of which 2 were ejected
4		3 of which 1 was ejected
5		1 was ejected
6		1 was ejected
7 & 8	1	14 of which 13 were ejected

The relative trajectory of the vehicle with regard to the pole was such that the head of the driver was crushed between the pole and the center-post of the vehicle (Severity Index of Lesions: 8).

Efficiency Of The Belt In Side Collisions

One hundred thirty-nine occupants were involved in side collisions and 28 of them wore belts. Figure 12 presents the synthetic distribution of this sample relative to the injury severity and the impact violence (ΔV in km/h). We also observe in this Figure (a) the place occupied relative to the impacted side of the vehicle, and (b) the presence or absence of an ejection. It is difficult to draw clear conclusions from the study of this figure since the belted occupants were not numerous.

We may, however, note with prudence some facts which have only an indicatory value.

1) *Moderate Violence Collision (ΔV 25 km/h).*

The belts were efficient for the occupants sitting on the side opposite the impact, because they prevented or reduced the severity of the projection against the inner side and they prevented ejection.

2) *Severe Violence Collisions (ΔV 26 km/h).*

The intrusion quickly became substantial and the belts did not prevent projection against the intrusion whatever the occupied places were. The belts once again prevented ejection.

Conclusions:

- The 3-point belts are efficient in frontal collisions (53% of all the accidents in Europe), in roll-overs (5.5%) and in some side collisions, that is to say in 70% of the total accidents.

CONDITION OF OCCUPANTS INVOLVED IN LATERAL IMPACTS, ACCORDING TO THE FORCE OF THE COLLISIONS

IMPACT SIDE		OPPOSITE SIDE
	FREE	●
□	EJECTED	■
	WITH SEATBELT	▲

139 OCCUPANTS (DRIVERS AND PASSENGERS)
INVOLVED IN 71 VEHICLES

SEVERITY OF THE INJURIES	VARIATION OF SPEED ΔV IN LATERAL COLLISIONS						
	I 5 to 15 km/h	II 16 to 25 km/h	III 26 to 35 km/h	IV 36 to 45 km/h	V 46 to 56 km/h	VI 56 to 65 km/h	TOTAL
FATAL VII and VIII		□□		● ▲▲	● ■■ ▲		19/4 E
VERY SERIOUS VI		■	□				3/2 E
CRITICAL V							—
DANGEROUS IV			●● ▲	■			5/1 E
SEVERE III			▲		●		6
MODERATE II	●● ▲	●●●●	●● ■ ▲		●		22/1 E
MINOR I	●●●●● ●●●● ▲▲	●●● ■ ▲▲	●●●●● ●●●●	● ▲▲			62/1 E
UNINJURED 0	●● ▲▲▲	●●●●	●●				22
TOTAL	39 (19 VEHICLES)	36 (18 VEHICLES)	41 (20 VEHICLES)	13 (8 VEHICLES)	9 (5 VEHICLES)	1 (1 VEHICLE)	139

Figure 12

- There is no case in which the belt was harmful.
- The efficiency of the belt was at its utmost in roll-overs mainly because it often prevented ejection (23%) in this type of accident and ejection increases the seriousness of the injury (about one out of two ejected is killed and the ejection is the direct cause of death in 87% of the cases).
- In frontal collisions, the efficiency of the protection afforded by the belt is statistically proved, including the most severe accidents (though it is stated that the sample *with* belts is involved in more severe collisions on the average than the sample *without* belts).
- The first fatal accident without belt occurs within the range of 16 to 25 km/h (ΔV) while it is necessary to reach a range *three times higher with a belt*.
- The probability of being seriously injured or killed is *six time lower with the belt* within ranges lower than a Δ V of 55 km/h, that is to say for 96% of all frontal collisions. One counts as a matter of fact:

Unbelted Occupants: eighteen dead and 8 very seriously injured, i.e. 5.5%

Belted Occupants: only *one* seriously injured person, and that injury was caused by the projection of a non-belted rear occupant, i.e. 0.9%

- The probability of being seriously injured or killed is *twice higher without a belt* for the most severe 4% accidents, despite factors then intervening to limit the efficiency of the belt.
- It is in the side collision that the belt efficiency is the lowest. It is still useful to prevent ejection and in collisions of moderate severity (ΔV lower than 25 km/h) to prevent or reduce the impact against the inner sides when the impact occurs on the side opposite the occupied places.
- The thorough analysis of technical causes of the injuries shows that the belt, always necessary, would still be more efficient with such complementary measures as:
  - general use of blocking retractors to eliminate statistically the looseness in the straps.
  - reinforcement of the structures to keep a sufficient deceleration volume in the occupant compartment in case of a very violent accident.
  - general use of belts for rear occupants.
  - reinforcement of seat hardware and anchorages to reduce the risk of overloading the belted occupants by impacts from occupants or luggage placed behind them.

The cost-benefit ratio of these measures should be calculated, to evaluate the judiciousness of their application.

FUTURE SAFETY STANDARDS AND THE ESV PROGRAM

SECTION FIVE PART 2

ESV – DEVELOPMENT AND LEGISLATION

DR. GÜNTHER BRENKEN, *VDA*
Des Verbandes Der Automobilindustrie E.V.

From the numerous reports presented by motor manufacturers at the 4th ESV Conference, it can be concluded that, after almost three years work on Experimental Safety Vehicles, a first development phase has largely been completed. Further intensive investigation is needed in the following areas:

- Concentration on biomechanical research
- Systematic evaluation of accident statistics to lay further foundations for the improvement, both directly and indirectly, of vehicle safety
- Development of better testing methods
- Creation of generally valid principles for multipurpose profit/cost analyses in the field of vehicle and traffic safety

The German motor industry has taken part at great expense in the evolution of the ESV and expects that new safety regulations will only emerge from the guaranteed results of research work on the ESV.

The German motor industry is prepared to cooperate with governments at any given time in setting up a universally applicable legislative framework in the field of vehicle mechanics, within which the results of the ESV studies can be put into practice. It would welcome the participation of manufacturers from all countries in this task. The aim must be to work out a long-term programme that finds recognition and can be put into use in Europe as well as the U.S.A. and Japan. In addition, all the other countries, who in the meantime still act according to their own rules, could bring themselves within this framework.

This suggestion by the German motor industry should serve to harmonize regulations in the field of vehicle technology or, in other words, the technology of safety. This would make long-term planning possible in which all motor manufacturers could join in due course.

The following example demonstrates the urgency of this request:

Since the Cincinnati court judgment the U.S.-Standard 208 has, in principle, been adhered to. Yet there are no dummies on which injury criteria can reliably be determined. Even if a new dummy-standard were shortly to be published as a *Proposed Rule*, the new equipment would only be ready after some considerable time. Extra months would then be needed to test and evaluate it. Only then would the authorities be in a position to work out the final legislation. At this stage only can the vehicle testing by the motor manufacturers follow, taking into account the new regulations.

With these facts in mind, the motor industry is faced with the question of which support systems should now be brought into operation in the U.S.A. and at what period in time. Since these or similar problems still exist in most countries, it seems to us that world-wide cooperation in this field is vitally necessary for the future.

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FUTURE SAFETY STANDARDS AND THE ESV PROGRAM

SECTION FIVE

PART 3

FUTURE SAFETY STANDARDS FOR JAPAN

HISASHI KAGEYAMA, *Director General*
Motor Vehicle Department
Ministry of Transport

THE JAPANESE ESV PROGRAM AT THE CURRENT TIME

On behalf of the host country, I wish to express our sincere appreciation to the delegates for their kind cooperation at this Fourth International Conference. Thanks to your consideration, the sessions are proceeding favorably and will end today with significant results. At this time, I would like to present our views on the Japanese ESV program and on safety standards for the future.

Work on the Japanese ESVs has been in progress in accordance with the requirements of the original program which was adopted in May, 1971. At that time, three automobile manufacturers, Toyota, Nissan and Honda agreed to participate in the ESV program. Test results and technical problems have been presented in detail by each company at earlier sessions. It is expected that through tests and further research, the remaining problems will be solved and final versions of each company's ESVs will be completed. Between the end of 1973 and 1974, ten cars from each company will be delivered to the Japanese government.

The final performance test of these ESVs will be conducted by the Japan Automobile Research Institute, Inc., for the Japanese Government. In making the performance test and in the assessment of the results, problems may arise with regard to the way in which test methods for assessment should be determined and implemented.

JAPANESE MOTOR VEHICLE SAFETY STANDARDS

At this time, I would like to briefly explain existing Japanese safety standards. An ordinance entitled "Safety Regulations for Vehicles Used for Road Transportation" (Ordinance No. 67) was issued by the Ministry of Transport in 1951. These Regulations were a consolidated version of the "Motor Vehicle Control Ordinance" (issued by the Ministry of Home Affairs in 1919), "Vehicle Regulations" (issued by the Ministry of Transport in 1947), and the "Law for Vehicles Used for Road Transportation" (No. 185 in 1951). The Regulations have been amended twenty-five times in the past twenty years to account for changes in traffic environment, for patterns in the use of motor vehicles, and for improvements in industrial technology. As a result, the contents have been greatly improved. The current Regulations provide for specifications, structure and function of motor vehicles in a broad and systematic way.

One of the conspicuous characteristics of the Regulations is that they contain many standards designed to prevent accidents involving pedestrians. This has been necessitated by the fact that a particularly large number of pedestrians in Japan have been killed or injured by motor vehicles. The requirements for the elimination of external projections on motor vehicles, the installation of undermirrors to detect the presence of obstacles immediately in front of motor vehicles, and equipment to prevent rear wheels from catching and injuring pedestrians are examples of these standards. In addition, seat belts, head restraints and the like have been required recently for passenger protection. However, the increasing number of fatalities will require further methods for reducing casualties resulting from traffic accidents.

SAFETY STANDARDS FOR THE FUTURE

It is necessary that motor vehicle safety standards be reenforced through the use of complete long term plans so that they will not be a myopic, symptomatic treatment of the problems we face. With this in mind, the Minister of Transport asked the Council of Transport Technology for recommendations concerning safety standards of the future. The Council consists of representatives of numerous fields. It includes professors, doctors, repair and maintenance people, traffic police, highway experts, newspaper editors and people from other related areas. As a result of the Minister's request, it was suggested that 63 standards be revised or added by 1976. (Copies of this recommendation have been distributed to the

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delegates.) In the future, Japanese safety standards will be enforced according to this recommendation.

In general, the following ideas should be adopted to insure motor vehicle safety. No structural or functional element of a vehicle should contribute to accidents or be the cause of an accident. (This may be referred to as a measure for avoiding accidents.) It should be noted, however, that regardless of what measures are taken to avoid accidents, it is impossible to completely eliminate poor driving procedures or mechanical failure of the vehicle. Indeed, in reality, it is impractical to aim at the complete prevention of accidents. However, in view of the fact that the number of drivers with relatively little experience has been increasing in recent years, measures designed to prevent accidents will become more important in the future. In addition, every effort should be made to devise safety regulations that will result in the greatest possible reduction of injuries both to passengers and pedestrians (measures to reduce casualties), and the prevention of fires (measures to prevent fires). Therefore, it is essential that the above mentioned measures be included in the motor vehicle safety standards.

Measures To Avoid Accidents

Legislation is needed to regulate the structure and functioning of motor vehicles. Such legislation should include:

1. Operating performance (acceleration/deceleration, braking efficiency, handling, stability, etc.),
2. Information-transmission systems (visibility, lights, switches, gauges, etc.), and
3. Structural strength (frame, body, etc.).

In addition, it is necessary to introduce the following requirements with due consideration for human engineering, medical science and psychology:

1. Prevention of poor driving procedures, and
2. Alleviation of driver fatigue (simpler handling, improvement of driver comfort, etc.).

Measures To Reduce Casualties

These consist of approximately the following two items:

1. External safety (reduction of casualties involving pedestrians and cyclists), and
2. Internal Safety (reduction of passenger injuries).

Fundamentally, the former should be determined in terms of the consolidation and improvement of the road environment as, for example, the separation of sidewalks from roadways. However, considering the numerous accidents involving injuries to pedestrians ("running weapon" accidents), to the greatest extent possible, safety measures should inlcude the regulation of the structure of motor vehicles.

As far as the latter is concerned, the number of accidents involving fatalities of vehicle occupants ("running coffin" accidents) is increasing. Greater safety in this area should be brought about by means of measures taken in the motor vehicle structure, in addition to the promotion of improved driver education and better road conditions. Thus, it is necessary to formulate safety standards for vehicle body strength and passenger protection devices. Such measures might be directed at causing no fatalities in vehicle collision in cases where vehicles are moving at designated low speeds.

Fire Prevention Measures

Needless to say, measures should be taken to prevent the outbreak of fires while vehicles are in operation. However, in view of the frequency with which fires have occurred as a result of collisions in recent years, it would appear that preventative measures are necessary in this area also.

Specifically, it is essential to prevent (1) the outbreak of fires involving electrical and fuel systems; and (2) the spreading of fires through the use of nonflammable material, fire-extinguishing equipment and the like. Moreover, careful consideration should also be given to the following:

1. Ease of Control (simplification of operation)
2. Increase in Reliability (certainty concerning operation and durability)
3. Ease in maintenance and inspection (ease in troubleshooting through inspection and repair work)
4. Fail-safe features in major components (structures should be such that the vehicle operates in the safest possible way in case of trouble)

Attention should also be given to the following items in order to implement this program:

1. Traffic Environment Requirements

   When motor vehicle safety standards are established, it is necessary to determine whether safety measures should be taken individually concerning the structure of motor vehicles, road design and driver education, or whether it is more effective to combine these for regulatory purposes. Further, it is essential to determine whether or not the standards to be enforced are designed to cope with social needs. In order to do this, opinions both of specialists and the public must be considered. In addition, an analysis of accidents must be made in order to ascertain the relationship between traffic accidents and the motor vehicle structure. Regulations should not be created as the result of random ideas. Moreover, even after the standards have been established, they should be subjected to periodic review so that they reflect the most recent accident analysis data and are consistent with changes in the traffic environment.

2. Guidelines for Technical Development
   Motor vehicle safety standards should be established so that they represent social needs and promote the necessary technical development. They should not be limited to what can be achieved under existing industrial technology.

3. Regulation of Performance
   As a rule, motor vehicle safety standards should avoid the specific regulation of structure and equipment so that the individuality of the various vehicles may be preserved. However, performance safety requirements should be regulated for all cars in general.

4. Internationalization of Testing Methods
   Since motor vehicles are international merchandise, safety standards and testing methods should be consistent with those adopted internationally, to facilitate comparison of performance and function.

The following items should be taken into account in order to enforce safety standards.

1. Appropriate Lead Time
   The lead time between establishment and enforcement of safety standards should be determined by considering the magnitude of social needs and the time required for technical development and mass production. Nevertheless, it is desirable that the time interval be as brief as possible.

2. Application to Motor Vehicles Already in Use
   Although it is desirable that the new motor vehicle safety standards be applied to motor vehicles currently in use to as great a degree as possible, such application should be based upon an assessment of social requirements, technical difficulties that may occur in the modification of vehicles, and possible disadvantages that may occur as a result of modifications.

SAFETY STANDARDS OF THE FUTURE AND FUTURE ESV PROGRAMS

As you will see, deliberations on the safety standard program and the ESV program began at almost the same time. Concepts concerning safety in the ESV program and the knowledge obtained from the creation of the Japanese ESVs are reflected in each respective area and stage.

It is our opinion that the knowledge acquired as a result of the ESV program should not be confined to ESV technology alone, but should be expanded and adapted for the production of vehicles to be sold on the open market.

In addition to the comments I have already made, I should like to add the following observations to the reports and presentations made by the delegates from other participating countries.

The ESV programs worked out up to now have given primary consideration to safety in the motor vehicle structure. In particular, passenger cars have been emphasized in terms of traffic conditions or human life. In other words, it seems that the extreme of safety in motor vehicle structures with special emphasis being placed on mechanical and structural technology has been pursued. Developments in this area should be carefully assessed.

Therefore, when the next ESV system or other safety plans for the future are considered, the position of the "motor vehicle" in terms of human life or at the very least, in terms of traffic systems, should be taken into account. Moreover, I feel that due consideration should be given to the following items although some of them have already been mentioned by other delegates at this conference.

1. Energy resources should be shared for transportation purposes.
2. Preventative measures should be taken to avoid the destruction of the environment as a result of highway and traffic conditions.
3. Measures should be enacted to insure that drivers of ESV vehicles are not careless as a result of the greater inherent safety of the cars which they operate.
4. Technology must be developed and performance requirements must be lowered, if necessary, so as to ensure compatibility of the ESV car of the future with existing vehicles (that is passenger cars, buses, trucks and other commercial vehicles), bicycles and pedestrians.
5. The amount of ground space available to motor vehicle traffic must be considered, as well as
6. Ease of repair in cases in which the ESV concept has been adapted to mass production.

In addition, cost-benefit analyses should be undertaken and should take the human element into account.

CONCLUSIONS

Clearly, the ESV program contributes to raising the level of automotive technology. Indeed, in support of this, it has been most gratifying to learn during this conference, that the world's motor vehicle manufacturers are conducting unhampered technological research and are engaging in the unrestricted development of safer motor vehicles. Programs such as these should be maintained, although it is necessary to make sure that technical development is not discouraged by impetuous legislation. However, when the knowledge obtained from this program proves to be useful in the real world and acceptable for human consumption, decision makers should take bold steps and have the courage and conviction to adopt the findings as law.

Considerable debate and decision making will be required to insure the success of future safety programs. Let us all meet the challenges that will be presented to us with increasing vigor and determination.

PROPOSED PROGRAM OF FUTURE SAFETY STANDARDS JAPAN

I Measures to avoid accident

Sept 1972
M.O.T., JAPAN

	Item	Classification				Contents	Application	Fiscal year of accomplishment					Remarks
		New item	Expanded item	Recommended item	Studying item			'72	'73	'74	'75	'76	
Field of Vision	1 1 Front visibility standard	0				To specify vision requirements of under mirror	Trucks (large size), buses	←→					It is necessary to examine the aspects of minimum vision, whether excessive vision is detrimental Japanese human engineering data etc
						To specify vision ranges in upper, lower, right and left areas of windshield	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks and buses		←—	—→			
	1 2 Windshield wiping area standard		0			To specify windshield wiping area			←—	—→			It is necessary to examine windshield wiper performance requirements relative to average rainfall in Japan
	1 3 Windshield washer standard		0			To specify technical standard of windshield washer		←→					
	1 4 Windshield defroster		0			To specify installation and technical standard of windshield defroster			←—	—→			It is necessary to study the requirements of large size windshields
	1 5 Headlight		0			To strengthen provisions of visible distance of passing beam and to specify passing beam headlight installation position		←→					
	1 6 Rearview mirror field of vision standard		0			Specify out and rear view fields of rear view mirror			←—	—→			It is necessary to examine the relation between required vision range and quality of image
	1 7 Rear window defogger			0		It is desirable to install rear window glass defogger							
Visibility, from exterior	2 1 Visibility of front light lamps		0			To strengthen provisions of visible distances of direction indicators and of clearance lamps and to specify the colours of these lamps	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks and buses	←→					

	Item	Classification				Contents	Application	Fiscal year of accomplishment					Remarks
		New item	Expanded item	Recommended item	Studying item			'72	'73	'74	'75	'76	
Visibility from exterior	I 9 Visibility of rear lamps		0			(1) To cause direction indicators to become independent and to specify their color (2) To strengthen provisions regarding visible distances of taillamps, stop lamps and turn signals (3) To strengthen provisions of luminous intensity ratio of combination stop lamps and taillamps	Ordinary passenger cars; small sized passenger cars; light passenger cars; multi purpose vehicles; ordinary trucks; small sized trucks; light trucks and buses	↔					
	I 10 Rearward visibility of trucks when loading or unloading	0				To install rear reflectors or the like which are visible when truck tail gates are lowered for loading or unloading	Ordinary trucks, small sized trucks and light trucks		←	→			When reflectors or the like are installed on the inside face of tailgates it is necessary to protect against breakage by cargo
	I 11 Side marker lamps	0				To specify installation of lamps or the like on both sides of vehicles	Large size trucks and trailers		↔				
	I 12 Emergency signal equip			0		It is desirable that advance warning triangle be provided for placing on the road to the rear of non moving vehicles to warn other vehicles of their presence							It is necessary to immediately specify performance and to take measures so that the equipment is properly used
Anti glare	I 13 Prevention of glare by headlight				0	To positively promote studies on the means to prevent drivers from being blinded by headlights of other vehicles (through improvement of light distribution characteristics, normalization of lighting angles under loaded and unloaded conditions, etc							
	I 14 Fog lamp performance standard	0				To specify fog lamp performance standard	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks and buses		↔				It is necessary to clarify the standard for use (to limit unnecessary lighting, for example)
	I 15 Anti glare of vehicle interior and exterior equipment	0				To specify places to be treated against glare and the limit on degree of reflection	Ordinary passenger cars, small sized passenger cars, light passenger cars multi purpose vehicles, ordinary trucks, small sized trucks, light trucks and buses		←	→			It is necessary to study the method of evaluation and the standard for judgment
Handling and stability	I 16 Handling and stability	0				To specify turning performance and slalom performance	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks and buses		←	→			It is necessary to study the method of evaluation and standard for judgment
						To upgrade above mentioned performance and to provide additional requirements				←	—	→	
Tires	I 17 Tire and rim selection	0				To specify appropriate combination between tires and rims	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks and buses		←	—	→		

	Item		Classification				Contents	Application	Fiscal year of accomplishment					Remarks
			New item	Expanded item	Recommended item	Studying item			'72	'73	'74	'75	'76	
Tires	I 18	Tire quality standard		0			To specify tire performance and of indication of wear limits	Ordinary passenger cars; small sized passenger cars; light passenger cars; multi purpose vehicles ordinary trucks; small sized trucks; light trucks and buses		↔	↔	↔		There are many matters to be examined regarding large sized vehicles.
	I 19	Safety of tires during blow out				0	To promote studies of such structure that prevents tires blow out when tires pick up nails, and that prevents the extreme loss of driving ability							It is necessary to carry out studies regarding safety of during tire
Brakes	I 20	Brake pedal depressing force and braking force		0			To strengthen provisions regarding pedal depressing force and to lower the upper limit	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks; light trucks and buses	↔					
							(1) To specify the relation between pedal depressing force and deceleration or braking force (2) To specify limits of fade of braking force when brakes are repeatedly applied			↔	↔			(1) It is necessary to examine the relations of braking capacity, stability and dual safety function. (2) It is necessary to examine the performance standard of respective vehicle types in accordance with actual conditions of use of vehicles.
	I 21	Behavior of vehicles when brakes are applied	0				To specify directional stability of vehicles when brakes are applied	··			↔	↔	↔	(1) It is necessary to examine the relation the performance standard of dual safety brakes (2) It is necessary to promote the study and of antiskid system
	I 22	Dual safety brakes		0			To strengthen provisions regarding installation of dual safety brakes	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles ordinary trucks, small sized trucks, light trucks and buses	↔					
							To specify braking capacity and warning system when part of dual safety brake system fails			↔	↔	↔		With tandem master cylinders, it is necessary to examine cylinder design and related increase in pedal stroke, in light car and cabover type vehicle
	I 23	Operating force of parking brake					To specify force required for operating parking brake	··	↔					
	I 24	Standard for brake hoses	0				To specify characteristics of brake hoses such as strength fatigue and aging	Ordinary passenger cars, small sized passenger cars, light passenger cars multi purpose vehicles ordinary trucks; small sized trucks; light trucks and buses		↔	↔			It is necessary to examine the relation with the periodic inspection and maintenance system
	I 25	Standard for brake fluid	0				To specify quality and characteristics of brake fluid	··		↔	↔			It is necessary to examine the relation with the periodic inspection and maintenance system

	Item	Classification				Contents	Application	Fiscal year of accomplishment					Remarks
		New item	Expanded item	Recommended item	Studying item			'72	'73	'74	'75	'76	
Brakes	1 26 Brake performance of vehicles combination	0				(1) To specify the delay time of braking effect (2) To specify directional stability of vehicles when brakes are applied	Vehicles combination		↔	↔			It is necessary to improve valves to provide proper response
Warning system	1 27 Overspeed warning system	0				(1) To specify the color of the speedmeter area above legal speed limit (2) To specify installation of audible warning equipment that indicates when legal speed limit is exceeded	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, small sized trucks, light trucks		↔				
	1 28 Lamp failure warningsystem			0		It is desirable to equip a system that enables the driver to identify when any safety lamp is inoperative							
	1 29 Brake fluid leakage warning system			0		To prevent accidents from brake malfunction it is desirable to develop a system that warns the driver when hydraulic pressure does not rise normally or when the level of brake fluid in the reservoir drops below the specified level							It is desirable that systems for advanced warning of trouble in brake systems be developed in the future
	1 30 Back up warning system performance standard	0				To specify the performance standard for back up warning systems (buzzer or horn)	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks, and buses		↔				It is necessary to specify the method of It is also necessary to examine whether to require the system in large sized vehicles
Control system	1 31 Indication of transmission select positions	0				To specify indication of transmission select positions	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks, and buses	↔					It is fundamentally necessary to standardize control positions
	1 32 Accelerator dual return system	0				To specify dual return system of accelerator linkage	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, small sized trucks, and trucks		↔				It is necessary to examine the relationship between increase in pedal depressing force and fatigue
Driver environment	1 33 Prevention of exhaust gas intrusion	0				To prevent exhaust gas from entering the compartment	Ordinary passenger cars, small sized passenger cars, light passenger cars, and multipurpose vehicles			↔	↔		
	1 34 Noise level in the compartment				0	To promote studies to lower the noise level in the compartment as much as possible							When external noise insulation is improved it becomes difficult to recognize external source audio signals

11 **Measures to reduce casualty**

	Item		Classification				Contents	Application	Fiscal year of accomplishment					Remarks
			New item	Ex panded item	Recom mended item	Study ing item			'72	'73	'74	'75	'76	
Injury prevention standard	11.1	Injury criteria				0	To promote studies on method of evaluation, and test for the purpose of establishing standards for prevention of injury of passengers in the case of collision as a total system of without unduly restricting individual equipment							
Strength of vehicle body	11.2	Strength of side doors	0				To specify strength and displacement of doors in side collisions	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, small sized trucks, and light trucks		←	→			Because the majority of vehicles in Japan are of small size it is necessary to examine the standard at that level
	11.3	Door retention system	0				To specify strength of door hinges and door locks	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, regular sized trucks, small sized trucks and light trucks		←	→			It is necessary to examine standard value of strength for each vehicle
	11.4	Reduction of amount of deformation into passenger compartment				0	To promote studies on method of evaluation and test, and standard for amount of deformation and intrusion into front, rear and side of passenger compartment for the purpose of maintaining survival space when vehicles collided, turn over or fall							
	11.5	Energy absorption of vehicle front and rear				0	To promote studies on method of evaluation and test, for the purpose of establishing standards for absorption of collision energy of vehicle front and rear end, so as to reduce the impact transferred to passengers during front or rear end collision							
	11.6	Seat [illegible]	0				To specify seat and [illegible] attached to [illegible]	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, and light trucks		←	→			It is difficult to apply this principle to folding seats, sideward facing seats and seats near emergency exits
							[illegible] for the purpose of establishing the standard for seat anchorage strength							There are many problems with buses such as strength of vehicle body
Passenger restraint system	11.7	Increase in number of seat belt installation		0			To strengthen the provisions to install seat belts and to have ordinary trucks equipped with seat belts at the outboard of front seats	Regular sized trucks		←→				
							To strengthen the provisions to install seat belts and to have vehicles equipped with seat belts at the outboard of rear seats	Ordinary passenger cars, small sized passenger cars, light passenger cars, and multi purpose vehicles		←→				(1) It is difficult to apply this principle to folding seats and sideward facing seats. (2) It is difficult to apply this principle to truck series station wagons and to cabover type passenger cars
							To promote studies relating to seat belt anchorages strength and seat belt requirements in buses							
	11.8	Lap and shoulder seat belts		0			To strengthen the provisions for installation of seat belts at outside of front seats and to require shoulder belt installation	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, small sized trucks, and light trucks	←	→				(3) It is necessary to study and develop seat belts for children (4) It is necessary to take measures to promote seat belt use

	Item	Classification				Contents	Application	Fiscal year of accomplishment					Remarks
		New item	Ex panded item	Recom mended item	Study ing item			'72	'73	'74	'75	'76	
Passenger restraint system	II 9 Seat belts with lock type retractors			0		It is desirable to use seat belts with lock type retractors for front outside seats because of easiness of use							It is necessary to further improve reliability
	II 10 Seat belts with warning system			0		It is desirable to install alarm equipment that gives warning to the driver to fasten his seat belt without fail							It is necessary to further examine operating con ditions of alarm
	II 11 Seat belts with interlock system			0		It is desirable to install such equipment that does not allow the engine to start unless the driver fastens his seat belt							
	II 12 Passive type passenger restraint system				0	To promote studies and development of passive type restraint system such as air bag and net type							There are many problems regarding reliability of equipment, medical aspect, effect in the case of multi collision and legislation (Law for con trol of High Pressure Gases, Law for control Explosives)
Interior impact protection	II 13 Increase of number of head restraint installed		0			To have head restraint installed at outboard of front seats and to strengthen the provi sions for installation of head restraint	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, small sized trucks, and light trucks	↔					Relation with rear visi bility is required to be studied
	II 14 Windshield glass				0	Since it is difficult to determine at this stage what kind of performance windshield glass should provide, examination with respect to the above will be promoted							
	II 15 Windshield mounting				0	Since mounting strength of windshield glass is closely related to the problem of II 14, it will be examined in parallel with the problem of II 14							
	II 16 Other window glasses		0			To specify the use of safety glass in ad dition to windshield for windows near passengers	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks, and buses	↔					
	II 17 Knock off standard of inside rear view mirror	0				To specify the stand ard for knock off of inside rearview mir ror in case of acci dents	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, small sized trucks and light trucks		↔				Examination of test method and evaluation method is required
	II 18 Shock absorp tion of seat back rear side (in cluding head restraint)	0				To specify shock ab sorbing standard of rear face of front seat back for pro tecting heads of rear seat passengers	Ordinary passenger cars, small sized passenger cars, light passenger cars, and multi purpose vehicles		↔				Examination of test method and evaluation method is required
	II 19 Shock absorp tion of instru ment panel	0				To specify shock ab sorbing standard of instrument panel for protecting heads of front seat passengers	Ordinary passenger cars, small sized passenger cars, light passenger cars, and multi purpose vehicles		↔				(1) It is necessary to ex amine the performance of independent instrument panels in relation to use and type of seat belts (2) Effect is limited due to the structure with vehi cles of cabover type

	Item	Classification				Contents	Application	Fiscal year of accomplishment					Remarks
		New item	Expanded item	Recommended item	Studying item			'72	'73	'74	'75	'76	
Interior impact protection	II 20 Control of projections inside passenger compartment	0				To specify that sun visors and arm rests should be free from sharp edges and projections	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, and light trucks		↔				
Measures for protection of pedestrians	II 21 Control of projections outside of passenger compartment	0				To specify that rear view mirrors, wheel spinners, wheel steps and spoilers should be free from sharp edges and projections	Ordinary passenger cars, small sized passenger cars, light passenger cars, multi purpose vehicles, ordinary trucks, small sized trucks, light trucks, and buses	↔					It is necessary to jointly examine both vehicle performance and pedestrian protection
						To specify that steps in cab width, bumper edges at side ends which may catch on pedestrian clothing and projections on rear body should be eliminated as a safety measure for pedestrians and oncoming vehicles	Ordinary trucks, small sized trucks		↔				
	II 22 Collapsibility of the outside mirror	0				To specify structures that flex, rotate or drop off in contact with a pedestrian during collision	Ordinary passenger cars, small sized passenger cars, light passenger cars, and multi purpose vehicles		↔				Structural vibration studies are required with cabover type passenger cars
						To promote research and development of outside mirrors for large sized vehicles for standardization of specification							It is necessary to examine weight of mirror and vibration of vehicle
	II 23 Side guards		0			To specify installation of side guards beneath side decks to prevent pedestrians from being caught by rear wheels	Ordinary trucks		↔				It is desirable that small sized trucks, if they are of high floor type, be equipped with side guard
	II 24 Shock reduction during pedestrian collision				0	To promote research and development to reduce injury of pedestrians during vehicle contact through softening of bumper surface material and improvement of structure of bonnet							(1) Effective soft bumper will require larger. Consequently the problem of vehicle size limits should be solved (2) It is necessary to examine this subject in regard to secondary collision with pedestrians
Measures to prevent damage to vehicles	II 25 Shock absorbing bumper				0	To examine installation and promote research and development of bumpers that absorb energy when impacted at low speeds							It is necessary to examine this subject together with standardization of bumper height
	II 26 Bumper height				0	To examine standardization of bumper height							It is necessary to establish an international height standard

III Measures to prevent fire

	Item	Classification				Contents	Application	Fiscal year of accomplishment					Remarks
		New item	Expanded item	Recommended item	Studying item			'72	'73	'74	'75	'76	
Prevention of fire	III 1 Fuel system fire prevention	0				To specify that there should be no fuel leakage from fuel systems such as the fuel tank and its pipe joints when the vehicle suddenly stops, when the vehicle makes a sharp turn or when the vehicle is in a collision	Ordinary passenger cars, small sized passenger cars, light passenger cars, and multi purpose vehicles		↔				It is necessary to examine both test and evaluation methods
	III 2 Electrical system safety				0	To promote research and development to prevent fire due to sparks at the time of collision							It is necessary to examine both test and evaluation methods
	III 3 Interior material non inflammability				0	To promote research and development to prevent rapid propagation of fire by making interior materials such as seat, roof and side panel linings fire retardant							(1) The problem of non inflammable materials producing noxious gases when exposed heat requires due consideration (2) Consideration is also required of durability of non inflammable treatment

Note 1 (1) "New item" is an item to be newly regulated in the future

(2) "Item to be expanded" is an item that should be reviewed or vehicle types for application should be expanded or contents of regulating should be modified and strengthened in the future, although regulations are partially applied today

(3) "Item to be recommended" is an item that is desired to be provided from the standpoint of safety

(4) "Item to be studied" is an item which investigation and study should be aggressively promoted in the future, technical development should be guided by and necessary measures should be taken in accordance with the results

2 "Multi purpose vehicle" means such a vehicle that is designed to transport passengers but that is also capable of transporting a limited cargo. A stationwagon is a typical example of a multi purpose vehicle

3 "Fiscal year to be accomplished" is the date by which technical data is collected and analyzed and resulting measures such as legislation and administrative guidance are established (lead time is not included)

SECTION SIX

CONCLUSION

PART I

CONCLUSIONS

HISASHI KAGEYAMA
Ministry of Transport
Japan

DR. GENE G. MANNELLA
United States Department of Transportation
United States

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SECTION SIX PART 1

CLOSING ADDRESS

HISASHI KAGEYAMA, *Director General*
Motor Vehicle Department
Ministry of Transport

As a citizen of the country which is hosting the Fourth International ESV Conference, it gives me great pleasure to be able to address you at this time. It is most gratifying to note that we have completed the entire program and, that this meeting has been extremely successful.

As you were able to see from the report which has just been given on Safety Standards of the Future, traffic accidents in Japan have been on the rise. In addition, exhaust emission, noise and other public hazards are beginning to reach serious proportions. It is my opinion that the ESV development project will contribute to vehicular safety and will be of assistance in combating the environmental problems created by the automobile.

It has been an honor for Japan to have held the Fourth International ESV Conference in Kyoto and to have been able to give all of you who come from so many different countries, a first-hand picture of traffic conditions here. Moreover, the conference has provided considerable inspiration to transportation administrators as well as to all people who are concerned with automobile production. As a result of this meeting, we are now even more committed to making a special effort to solve safety problems and to promote international methods of dealing with these difficulties in the future.

In closing, permit me to extend special thanks to the United States Government for making it possible to hold this conference. At the same time, I should also like to express sincere appreciation for the valuable reports presented by delegates from the United States, Germany, the United Kingdom, Italy, France, Sweden, Belgium, and the Netherlands.

We of Japan are most grateful to all of you who have come so far in order to participate in this conference. I look forward to seeing you next year at the Fifth International ESV Conference which will be held in London.

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SECTION SIX PART 1

CONCLUDING REMARKS

DR. GENE G. MANNELLA, *Associate Administrator for Research and Development*
Research Institute
National Highway Traffic Safety Administration

I will make my concluding remarks quite brief, since most of the sentiments that I feel have been expressed quite well.

One comment I would like to make concerns the conference itself. I am impressed with the delegates' sincerity and level of concern for the very significant societal problem that road safety represents.

Although differences of opinion in approach were represented here, I believe that such honest dissensions are constructive. I also believe that the most lasting contribution that these conferences can achieve is to improve and enhance our willingness to discuss and resolve these differences of opinion. With this in mind, I turn your attention to London in June of next year and a further opportunity that we will have to exchange ideas and technical data on achieving improved automotive safety.

I now declare this Fourth International Technical Conference on Experimental Safety Vehicles closed.

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FOURTH
INTERNATIONAL
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CONFERENCE
ON EXPERIMENTAL
SAFETY
VEHICLES

MARCH 13-16
1973

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Under the ESV Project of the CCMS Road Safety Pilot Study, automobiles are being designed with safety as the primary criteria. The principal objectives are to:

- Demonstrate the feasibility and performance of new safety features.
- Develop techniques for designing safety into a vehicle as a whole rather than as "add-on's."
- Stimulate the development of new technology in vehicle safety.
- Promote international cooperation in developing and sharing new auto safety technology.

The major safety performance requirements are in the areas of:

- *Crash Injury Reductions* – The ESV must protect occupants from death or serious injury at the equivalent of a 40-50 mph head-on crash into a fixed barrier, side pole impact, and rollover, without presenting unnecessary hazards to other automobiles or pedestrians.
- *Accident Avoidance* – The ESV must meet stringent specifications on braking, steering, handling, lighting, visibility, and other features which will aid the driver in avoiding crashes.

Because the frightful toll of deaths and injuries from motor vehicle accidents is a social problem of increasing concern in every nation of the world, the United States is leading a Pilot Study in this area. The overall program includes a number of individual projects led by a NATO ally, together with other NATO and non-NATO countries, participating in varying degrees. The international ESV effort is one of these projects. Other study projects are Accident Investigation (The Netherlands), Alcohol and Highway Safety (Canada), Motor Vehicle Inspection (Germany), Emergency Medical Services (Italy), Pedestrian Safety (Belgium), Road Hazards (France).

The Federal Republic of Germany, Italy, France, the United States, and the United Kingdom are all actively working on the development of ESV's. Two non-NATO countries, Japan and Sweden, are also active partners in this safety-vehicle effort. All participants are kept informed of progress through exchange visits by teams of engineers and through open discussions at public meetings held in conjunction with the CCMS program. The first of the major international meetings on ESV development was held in Paris, January 1971; the second in Stuttgart in October 1971; the third in Washington in May, 1972. The fourth conference was held in Kyoto, Japan in March of 1973. The fifth conference is scheduled for England during the Spring of 1974.

In the United States, the lead country for the ESV project, contracts were awarded to two non-automobile companies, Fairchild Industries and AMF Incorporated. Following this, General Motors Corporation and the Ford Motor Company each joined the program under a "one-dollar" contract to develop their own ESV's. All four companies have developed prototype vehicles of approximately 2,260 kg (5,000 pounds) weight to meet high levels of safety performance.

Japan has subsidized ESV research and three of its major automobile manufacturers have large investments in the development of ESV's. Nissan (Datsun) is producing ESV prototypes of 1,200kg (2,600 pounds) weight range; Honda in the 820kg (1,800 pounds) category; and Toyota in the 900kg (2,000 pounds) category.

In the Federal Republic of Germany, Mercedes-Benz has created ESV models in the 2,100kg (4,600 pounds) range with restraint and energy-absorbing structure combined with some of the convenience features and styling of their production vehicles. Volkswagen has developed their ESVW in the 1,450kg (3,200 pounds) range with consideration of cost effectiveness and the problem of aggressivity, i.e., the problem of large vs. small cars in a collision. Opel has conducted studies and tests of the energy and forces involved in automotive collisions.

The United Kingdom, until recently, has concentrated on the development of systems and subsystems by the automobile manufacturers and their suppliers. Some of their systems have been incorporated by

British Leyland into ESV models of the MGB and Triumph. The UK is starting construction on a new ESV vehicle with particular emphasis on aggressivity.

Italian companies participating in the ESV program of Italy are Fiat and Alfa Romeo. Fiat has developed three models of an ESV in the weight ranges of 680kg (1,500 pounds), 820kg (1,800 pounds) and 1,000kg (2,200 pounds). The small Fiat ESV has been crash tested in the United States in order to obtain insight into the problem of large versus small car collision damage.

France, in cooperation with Citroen, Peugeot-Renault, is supporting the ESV program. The French Experimental Safety Sub-System (ESSS) Program is studying the safety problem from the subsystems level before entering into development of a complete ESV.

In Sweden, the Swedish Board for Technical Development and Saab-Scania and Volvo are working on various portions of the ESV program. Volvo has developed its own versions of the Volvo Experimental Safety Car (VESC) of 1,450kg (3,200 pounds) curb weight.

The ESV program has involved the major automobile manufacturers of the world in this cooperative effort to develop safety cars. It has fostered an international exchange of information on vehicle safety design, raised the awareness of governments and the public of the possibilities for producing safer cars and brought the day closer when consumers can purchase cars which will be far safer than current models.

As the body of knowledge builds up from these ESV developments and tests, work will proceed on drafting the regulations for similar performance requirements of production vehicles, taking into account the time necessary to adapt the research findings into production vehicles at prices people can afford.

ESV EXHIBITION

During the Conference Period the following ESV prototype and components were displayed at the Conference Hall.

ESV Prototypes:

Daimler-Benz ESF 22	West Germany
Fiat 1,500, 2,000 & 2,500 ESV	Italy
General Motors ESV	U. S. A.
Honda ESV	Japan
Nissan ESV	Japan
Toyota ESV	Japan
Volkswagen ESVW	West Germany

Components and Others:

Bosch	(Components)	West Germany
Nihon Seiko	(Components)	Japan
Takata Kojyo	(Seat belts)	Japan
Transports and Road Research Laboratory	(Poster boards and dummy)	United Kingdom

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NISSAN
ESV
E2

MB-ESF 22
22
for the Future

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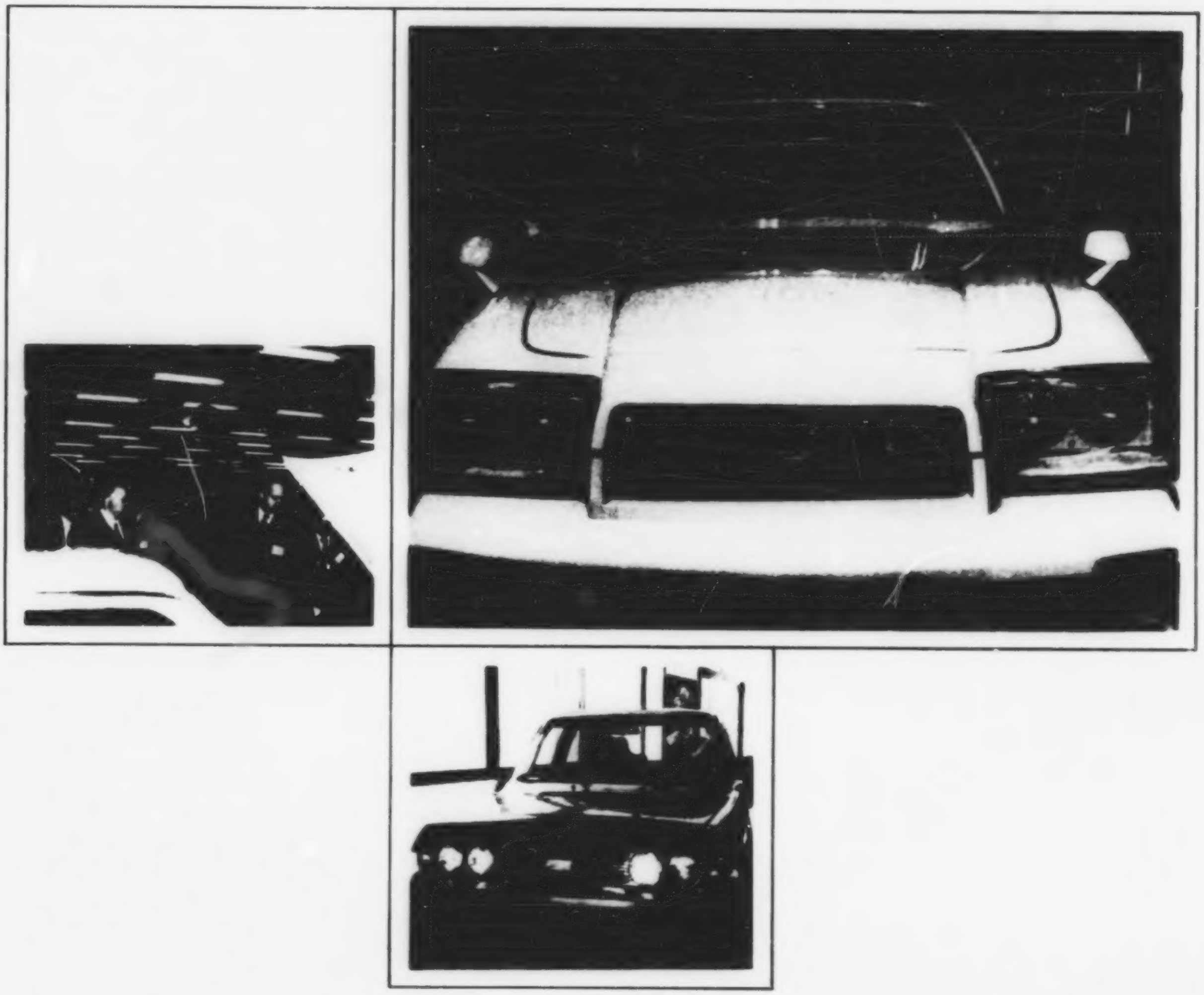
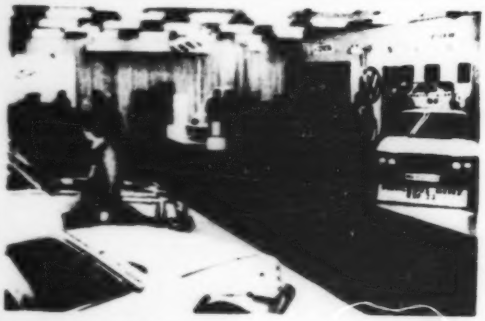
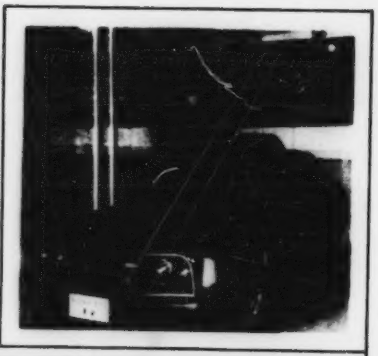


MERCEDES-BENZ
ESF 22
Engineering Safety for the Future

NISSAN ESV
E 2



ESVW



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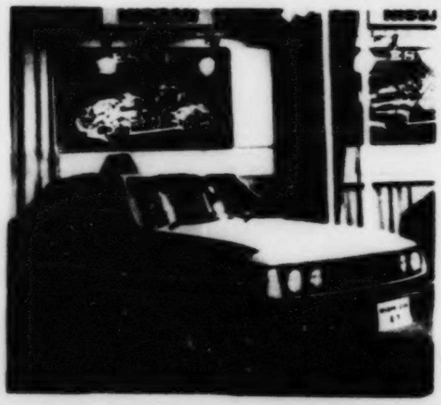

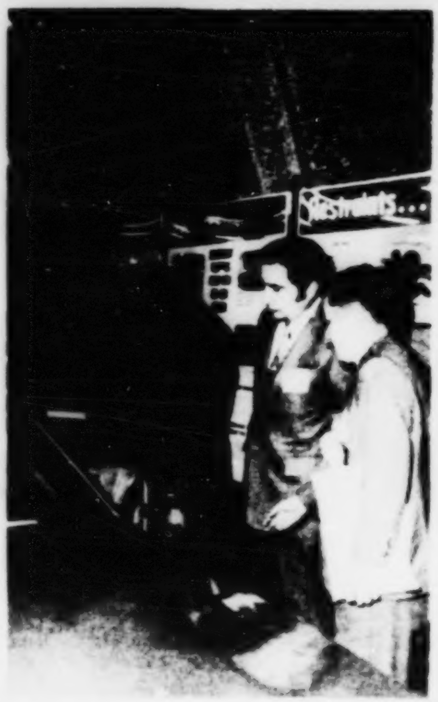

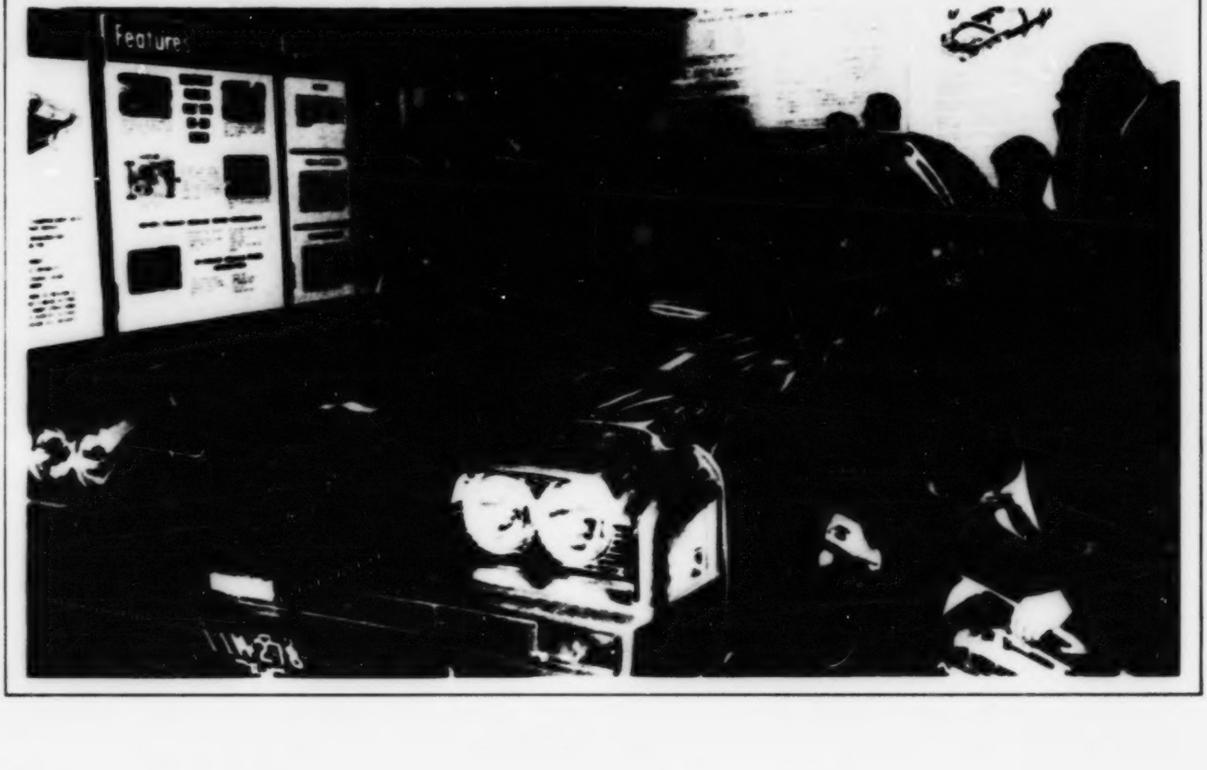
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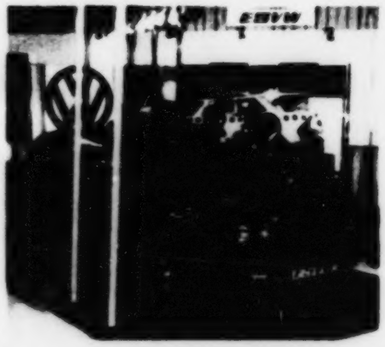

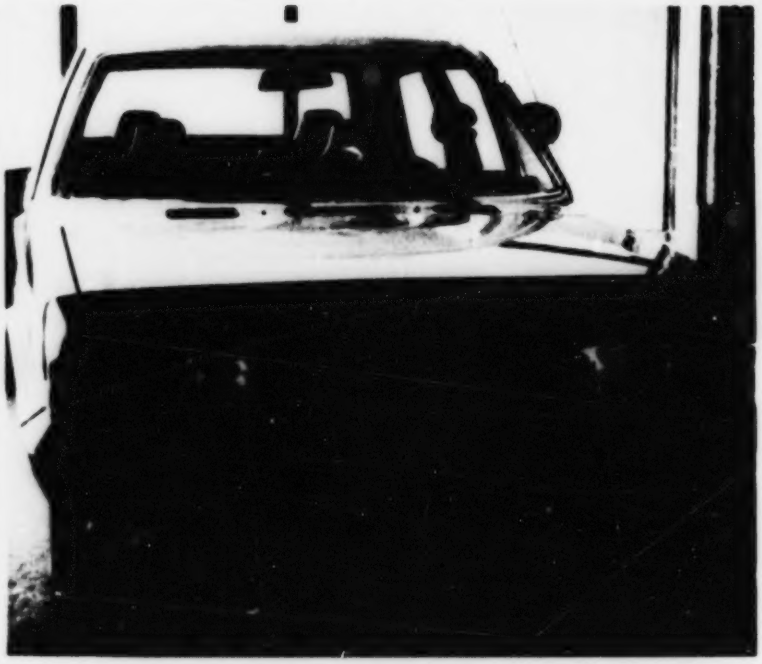

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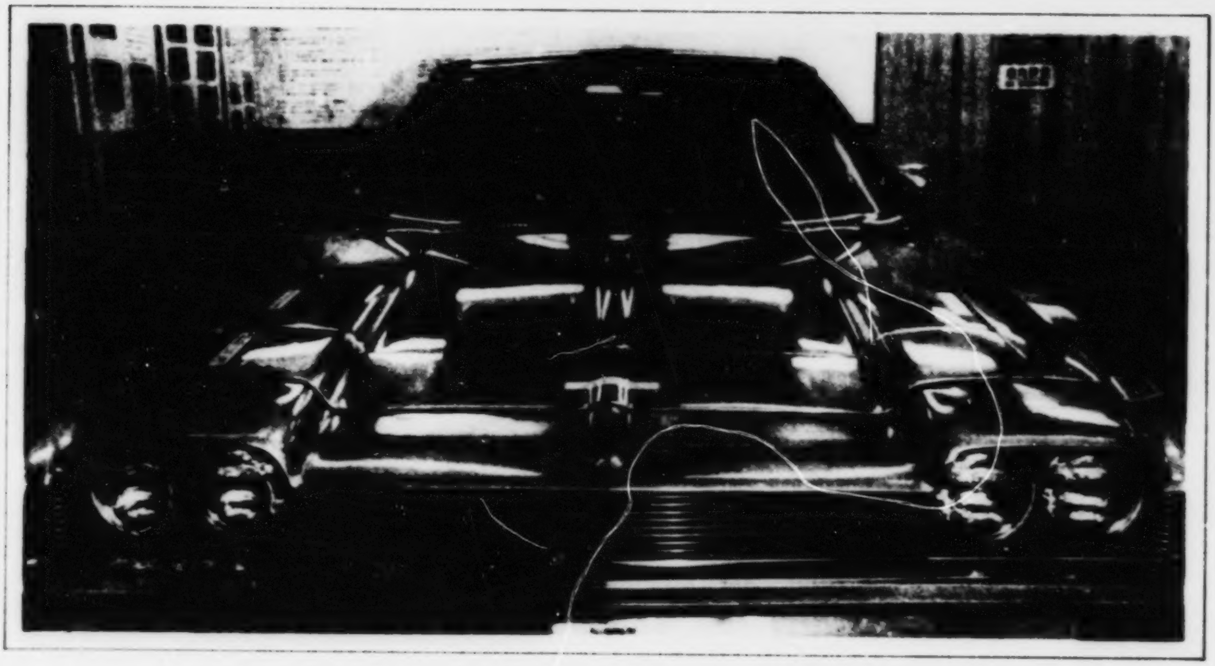

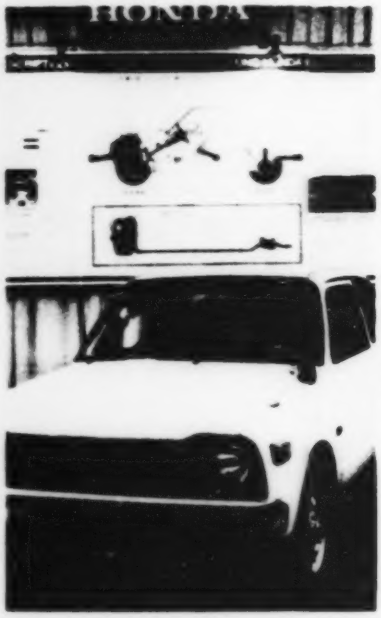
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